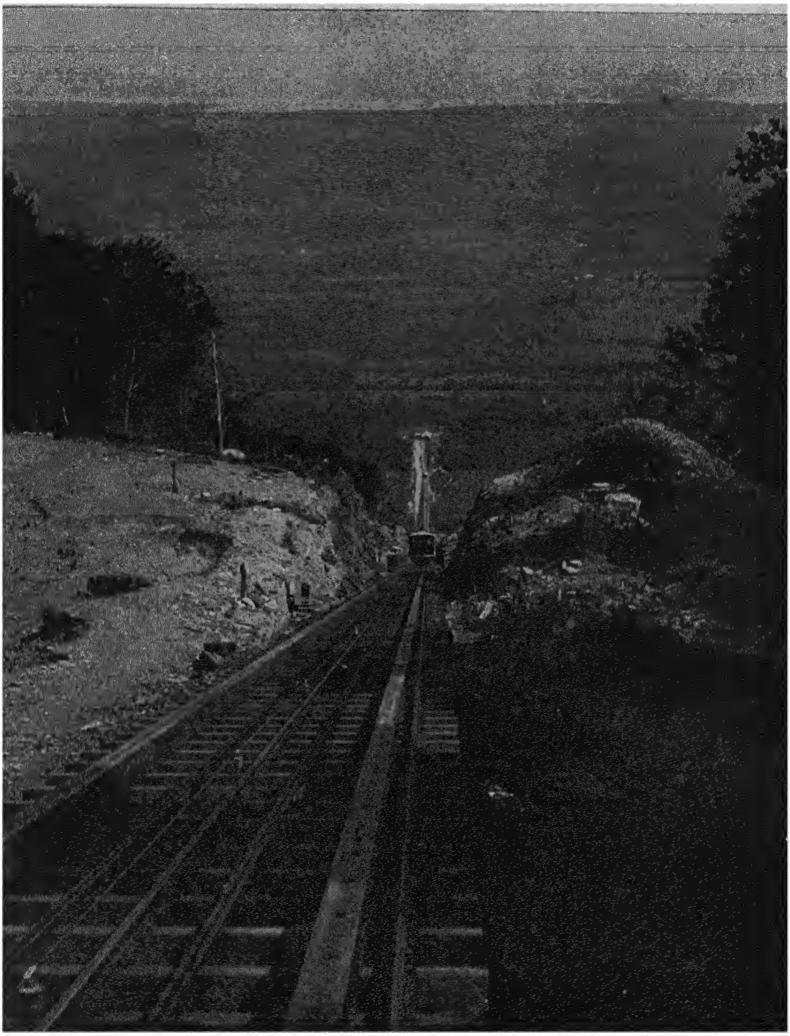
DAMAGE BOOK



**VIEW OF THE CATSKILL MOUNTAIN INCLINE RAILWAY**

The total length of the road is 7000 feet, with an average grade of 23 per cent. The road is of the three-rail type and as usual in this type of railway the two cars are attached to the same cable and form a partially balanced system. Safety devices are provided in case the cable breaks.

*Courtesy of Otis Elevator Company*

# Cyclopedia

*of*

# Engineering

*A General Reference Work on*

STEAM BOILERS AND PUMPS; STEAM, STATIONARY, LOCOMOTIVE, AND MARINE ENGINES; STEAM TURBINES; GAS AND OIL ENGINES; GAS-PRODUCERS; COMPRESSED AIR; REFRIGERATION; ELEVATORS; HEATING AND VENTILATION; MANAGEMENT OF DYNAMO-ELECTRIC MACHINERY; POWER STATIONS; ETC.

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*Assisted by*

CONSULTING ENGINEERS, TECHNICAL EXPERTS, AND DESIGNERS OF THE HIGHEST PROFESSIONAL STANDING

*Illustrated with over Two Thousand Engravings*

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Grateful acknowledgment is made here also for the invaluable co-operation of the foremost engineering firms in making these volumes thoroughly representative of the best and latest practice in the design and construction of steam and electrical machines; also for the valuable drawings and data, suggestions, criticisms, and other courtesies.

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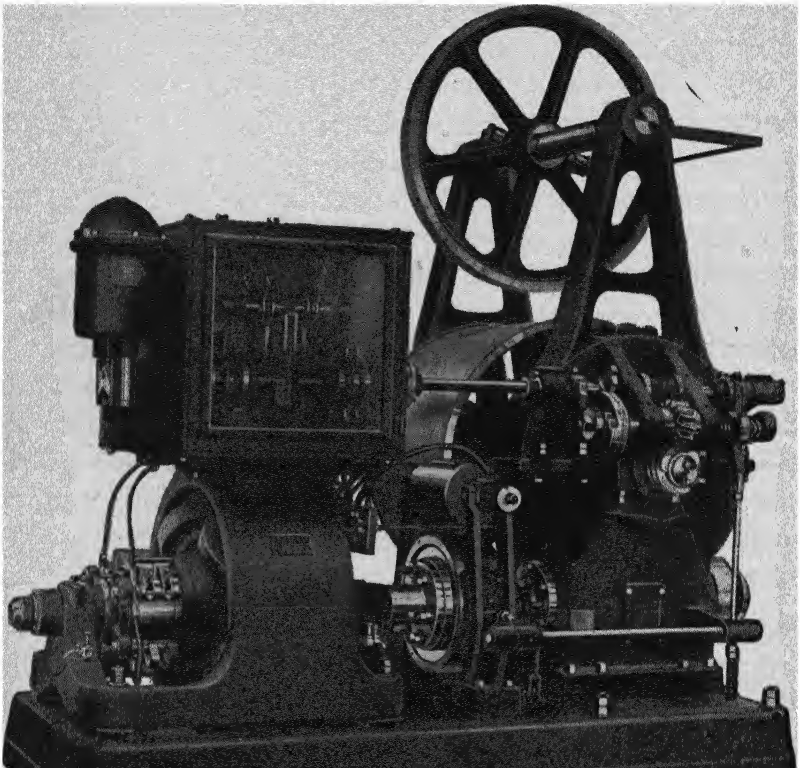

HEAVY PASSENGER ELEVATOR ENGINE WITH FULL ELECTRICAL CONTROL
*Courtesy of Warner Elevator Manufacturing Company, Cincinnati, Ohio*

# Foreword

THE "prime mover", whether it be a massive, majestic Corliss, a rapidly rotating steam turbine, or an iron "greyhound" drawing the Limited, is a work of mechanical art which commands the admiration of everyone. And yet, the complicated mechanisms are so efficiently designed and everything works so noiselessly, that we lose sight of the wonderful theoretical and mechanical development which was necessary to bring these machines to their present state of perfection. Notwithstanding the genius of Watt, which was so great that his basic conception of the steam engine and many of his inventions in connection with it exist today practically as he gave them to the world over a hundred years ago, yet the mechanics of his time could not build engine cylinders nearer true than three-eighths of an inch — the error in the modern engine cylinders must not be greater than two-thousandths of an inch.

¶ But the developments did not stop with Watt. The little refinements brought about by the careful study of the theory of the heat engine; the reduction in heat losses; the use of superheated steam; the idea of compound expansion; the development of the Stephenson and Walschaert valve gears — all have contributed toward making the steam engine almost mechanically perfect and as efficient as is inherently possible.

¶ The development of the steam turbine within recent years has opened up a new field of engineering, and the adoption of this form of prime mover in so many stationary plants like the immense Fisk Station of the Commonwealth Edison Company, as well as its use on the gigantic ocean liners like the Lusitania, makes this angle of steam engineering of especial interest.

¶ Adding to this the wonderful advance in the gas engine field — not only in the automobile type where requirements of lightness, speed, and reliability under trying conditions have developed a most perfect mechanism, but in the stationary type which has so many fields of application in competition with its steam-driven brother as well as in fields where the latter can not be of service — you have a brief survey of the almost unprecedented development in this most fascinating branch of Engineering.

¶ This story has been developed in these volumes from the historical standpoint and along sound theoretical and practical lines. It is absorbingly interesting and instructive to the stationary engineer and also to all who wish to follow modern engineering development. The formulas of higher mathematics have been avoided as far as possible, and every care has been exercised to elucidate the text by abundant and appropriate illustrations.

¶ The Cyclopedia has been compiled with the idea of making it a work thoroughly technical, yet easily comprehendible by the man who has but little time in which to acquaint himself with the fundamental branches of practical engineering. If, therefore, it should benefit any of the large number of workers who need, yet lack, technical training, the publishers will feel that its mission has been accomplished.

¶ Grateful acknowledgment is due the corps of authors and collaborators — engineers and designers of wide practical experience, and teachers of well-recognized ability — without whose co-operation this work would have been impossible.

# Table of Contents

## VOLUME VI



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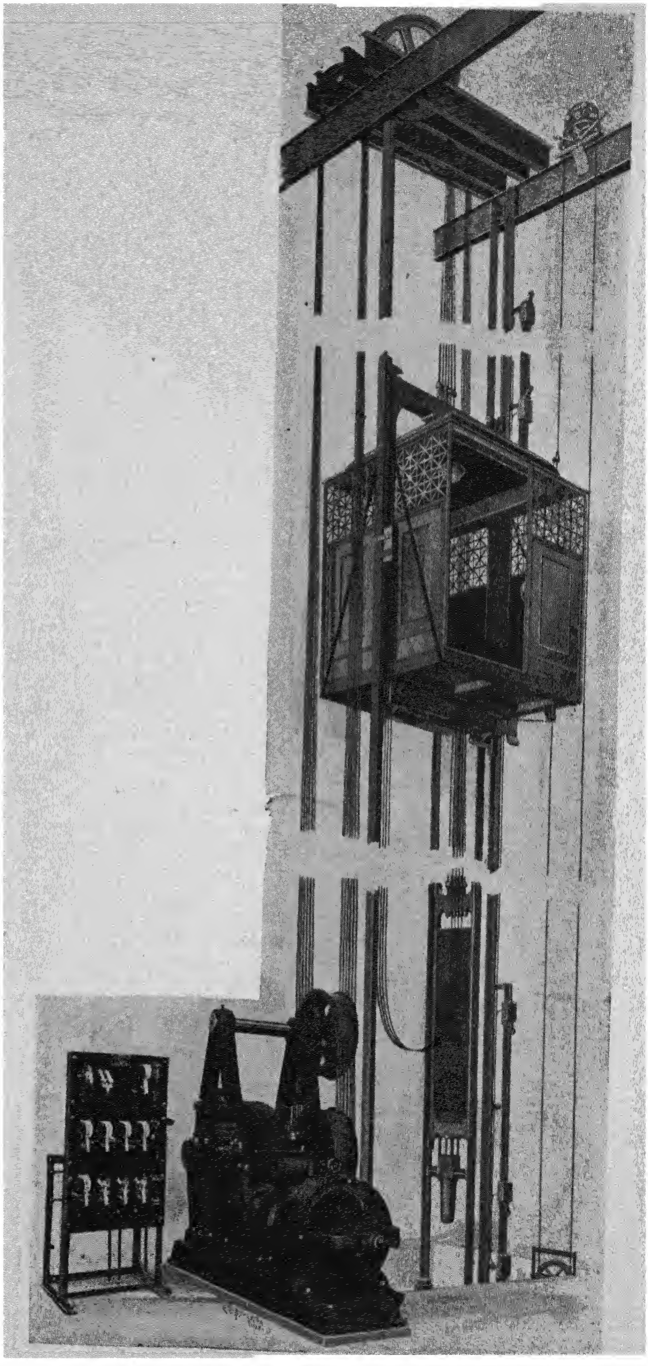
* For page numbers, see foot of pages.

† For professional standing of authors, see list of Authors and Collaborators at front of volume.



LAYOUT OF HOUGHTON TANDEM-GEAR TRACTION ELEVATOR INSTALLATION

*Courtesy of Houghton Elevator and Machine Company, Toledo, Ohio*

# ELEVATORS

## PART I

## HAND POWER ELEVATORS

### HISTORICAL DEVELOPMENT

**Fundamental Principles.** The development of the elevator to its present state of efficiency has been very gradual, and in this respect it does not differ from the majority of those inventions which have become factors in modern civilization. The stationary steam engine, the locomotive, and the steamboat are instances of inventions which, starting in a very humble way, have by their utility materially assisted in the advancement of business and commerce, and which, in turn, have themselves been developed to meet the increasing demands made upon them.

Undoubtedly the hand elevator, as it is called, or, to be more explicit, the lifting machine operated by manual power, was in existence in one form or another for centuries before the modern appliance with its traveling platform or cage was thought of. The Chinese windlass, the rope tackle, the capstan, and the ship windlass are all primitive forms of the lifting machine which depends on manual energy for its operation, and there is no doubt that the same necessities which led to the invention of these simpler forms of lifts were responsible for later developments.

### SLING TYPE

**General Construction.** Lack of space prevents our going into a lengthy description of the earliest types, and we shall begin with the elevator as it existed some fifty or sixty years ago. At that time the hand elevator was to be found in nearly all warehouses of two or more stories in the form of the "sling machine", so called because of the absence of any platform or cage. The load of goods to be lifted was secured in a strap—or sling—of rope which, in turn, was attached

to the hoisting rope of the hand lift, by means of which the load was elevated to the upper story or loft of the building, Fig. 1.

**Drum.** The apparatus as shown in Fig. 2 included a wood drum or roller about 11 inches in diameter for winding up the hoisting

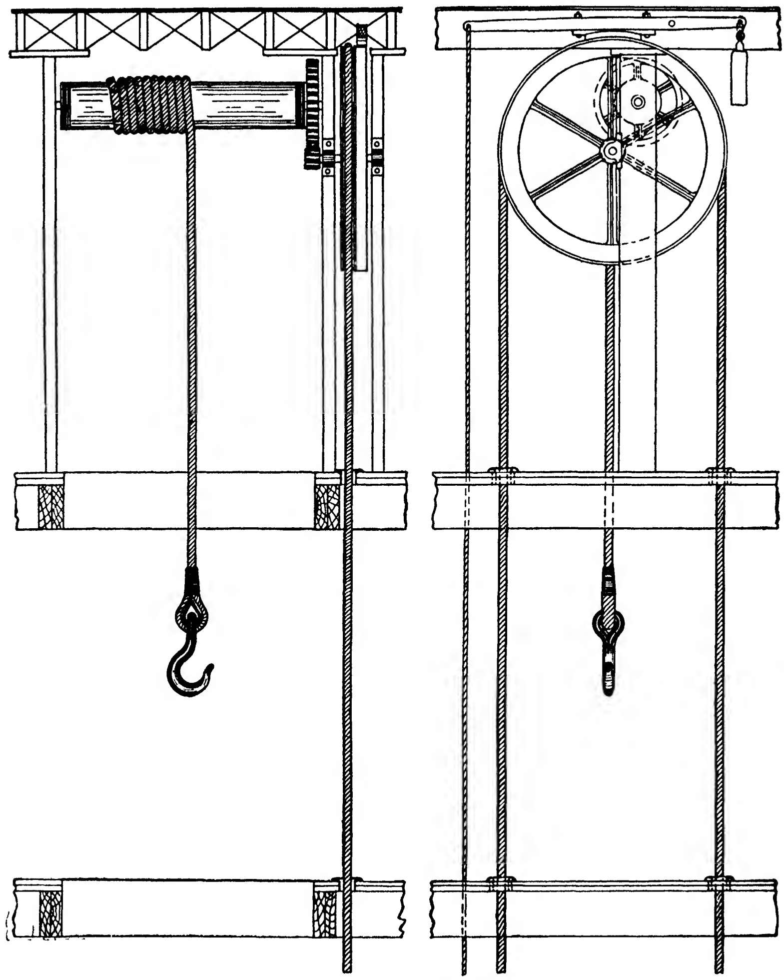


Fig 1. Front and Side Elevation of Sling Type of Hand Elevator

rope. This drum had a spur gear securely attached to one end of it by staples, and was provided at the ends with gudgeons, or journals, which revolved in babbitted boxes or bearings. The purchase was obtained by means of a spur pinion meshing with the teeth of the

gear and a large rope wheel or sheave, both mounted on a short shaft and keyed securely thereto. This shaft was also provided with journals and boxes, the whole being set up in a framework made of wood and securely attached to the building frame in the loft.

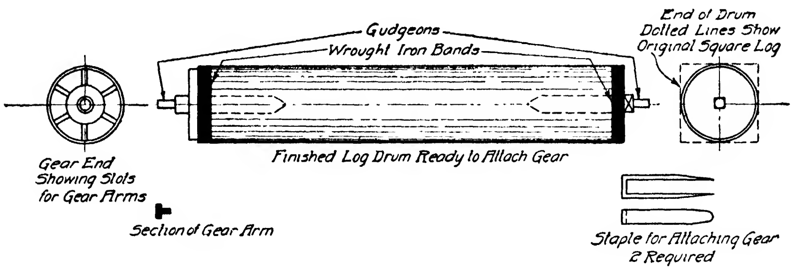



Fig. 2. Details of Finished Log Drum

Below the machine was a "hatchway" or series of floor openings located directly above one another, through which merchandise in bulk was to be handled. The drum was placed directly over the hatchway and the hoist rope dropped down through the opening, the free end

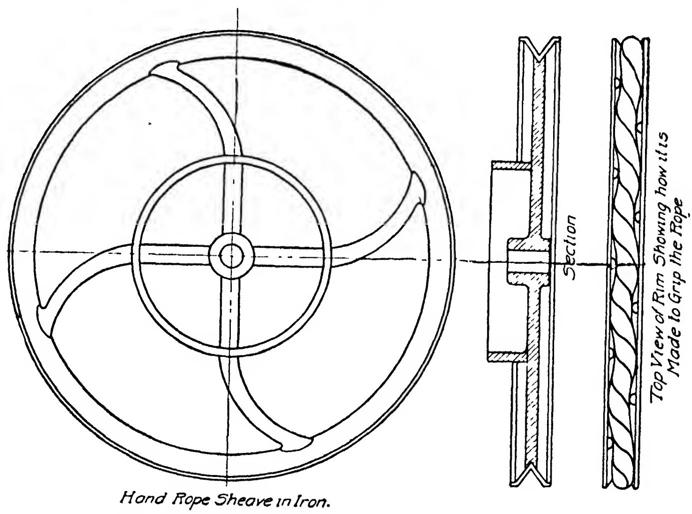



Fig. 3. Elevation, Plan, and Section of Rope Wheel with Ridges Cast in Score

of the rope being provided with a strong wrought-iron hook for attaching to the load to be lifted.

**Gearing and Rope Wheel.** The ratio of pinion to gear was usually about 5 to 1, the diameter of the large rope sheave was from

4 to 5 feet, and the groove or channel in the rope wheel was V-shaped to prevent the rope from slipping. As a further precaution, small ridges were cast in the score, the ridges being about 12 inches apart and alternating from one side of the channel to the other. This arrangement caused the rope to lie in the channel in a serpentine form, thus effectually preventing slippage, Fig. 3.

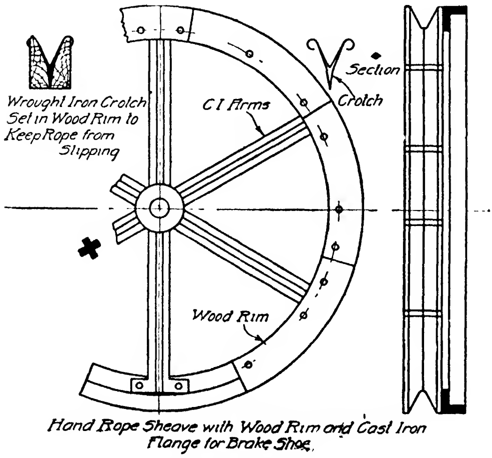



Fig. 4 Details of Rope Wheel with Cast-Iron Arms and Hub

In many instances the large rope wheel, instead of being made entirely of iron, had a cast-iron hub with iron spokes

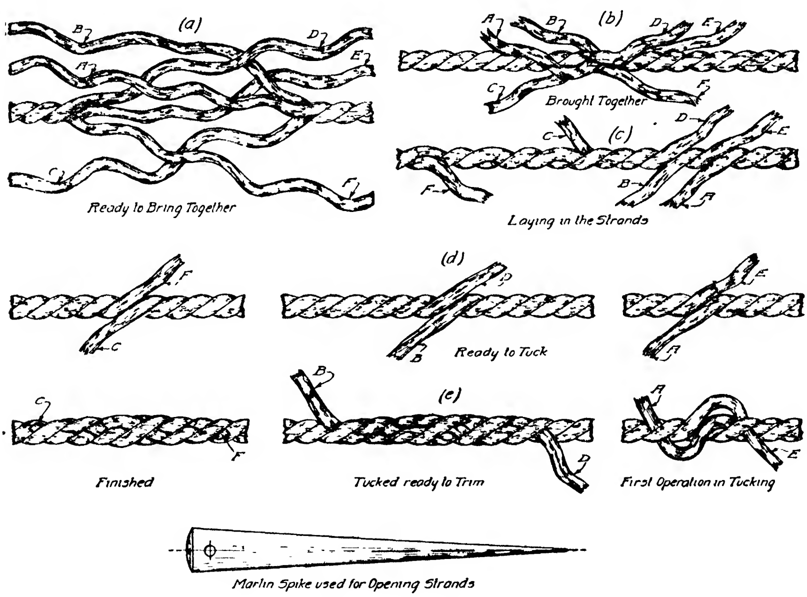



Fig. 5. Diagram Showing Method of Long-Splicing Rope

and a rim of wood made of segments sawed from 2-inch dressed plank nailed and bolted together; a V-shaped groove was turned in the rim

or worked out by hand during the process of building up the rim, Fig. 4.

**Tackle.** The rope used was usually of manila, $1\frac{1}{4}$ inches in diameter, and made endless by means of a "long splice" as shown in Fig. 5, yarns being removed from the rope ends during the process. Before being spliced, the rope was passed or "riven" through holes in the floors of the building, the position of the holes being determined by plumbing down from the large rope wheel after it was in position. Cast-iron thimbles were used to protect the holes from wear, as shown in Fig. 1.

The hoisting rope was usually of $1\frac{1}{2}$-inch diameter, the fixed end being attached to the drum by means of three or four large iron staples made from $\frac{3}{8}$- or $\frac{7}{16}$-inch round iron, spanning the rope and driven into the wood drum at intervals of 5 or 6 inches. The rope was made long enough to reach the lowest floor and still leave several turns around the drum, the spare turns having as much to do with the security of the hoist rope as the staples.

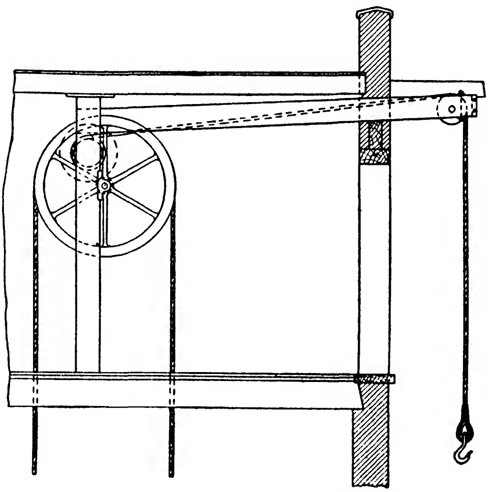


Fig. 6. Details of Outrigger Arrangement for Sling Hand Elevator

The ratio of purchase in a machine such as this was about 25 to 1, the gear and pinion ratio being 5 to 1, and that between drum and rope wheel 5 to 1. Two men, therefore, could easily lift 800 to 1000 pounds two or three stories without undue fatigue, and nearly double that load upon emergency.

**Lowering Brake.** The lowering of loads was accomplished by means of a brake either in the form of a shoe applied to the rim of the large rope sheave, or of a yoke or a band applied to a pulley cast or bolted on the arms of the wheel. This will be described further on.

**Outrigger.** In many instances, instead of using a hatchway inside the building, an outrigger was employed, consisting of a heavy

beam made of two stout planks bolted together with sufficient blocking between to leave a space of four or five inches in which a large iron sheave was placed near one end for carrying the hoisting rope, Fig. 6. This beam was placed close up under the roof of the building, the sheave-end projecting through the outer wall about three feet. By having a wide doorway at each story directly opposite the lifting rope, goods could be lifted from wagons directly into the building.

The writer has gone somewhat into detail in describing this machine because it is the type from which the hand-power elevator of today has developed, and a clear conception of its features will enable the student to understand the later types.

## DEVELOPMENT AND PRINCIPLES OF THE MODERN HAND TYPE

**Differences Between Modern and Sling Types.** The modern type of hand elevator differs from the one just described in the following particulars:

(1) In the use of a platform or traveling car on which to load the goods to be handled.

(2) In the use of guide rails for the car to travel on.

(3) In the use of antifriction roller bearings to reduce friction in operation.

(4) In the application of a heavy weight to counterbalance the weight of the car.

(5) In the use of wire ropes or cables of smaller diameter and greater flexibility than the old cumbersome manila lifting rope.

(6) In the addition of safety appliance to hold the car from falling in case of the cables breaking.

(7) In the use of an improved form of brake for stopping and holding the car rigidly at any floor while loading and unloading.

These features of the modern type of hand-power elevator will now be discussed in the order given.

These hand-power elevators are made and used today very extensively, finding their greatest field of application in small stores for the purpose of handling goods between the basement and upper floors. Low first cost and simplicity of construction and operation are their chief recommendations, and they fill all the requirements of

an elevator where the number of loads to be lifted per day is not excessive; however, care is absolutely necessary in their operation. In the first place, they should not be run unchecked—either down with a load or up without one; and in the second place, they should be securely locked when stopped for loading or unloading. If these precautions are taken, and if proper attention is paid to lubrication, they are both serviceable and durable.

**Introduction of Platform.** A very important step in the development of the elevator was the introduction of the platform or traveling car, Fig. 7, and a new field for its application was thereby created. Formerly the elevator was employed for handling only goods in bulk, but the introduction of the platform or car made it serviceable in retail stores and other places where it was desirable to hoist or lower smaller packages in quantities.

*Necessity for Lightness.* Of course it soon became apparent that lightness in weight was a feature which could not be ignored, for, even though the platform may be well counterbalanced, the addition of a counterbalancing weight adds to the

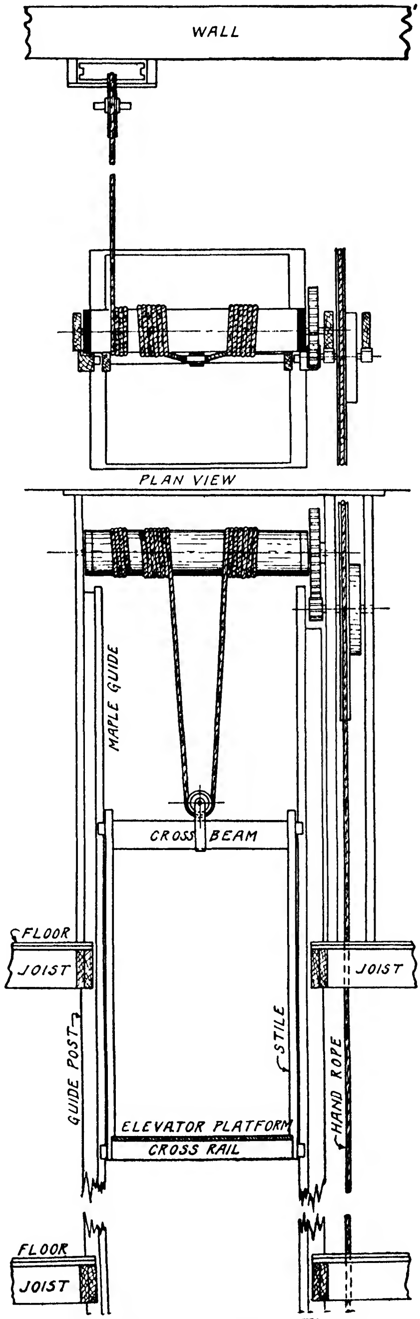



Fig. 7. Evolution of Platform Elevator from Sling Machine

friction in operation, to the inertia in starting, and to the momentum in stopping, thus creating forces which have to be overcome by the operator. Hence platforms for hand elevators are always made as light as may be consistent with the necessary qualities—strength and durability.

**Guidepost.** Naturally the guidepost and a rail at each side of the hatchway came into use simultaneously with the platform. The first type of rail used was made of cast iron with a flange for attaching it to the guidepost by means of wood screws and with a serrated edge for engaging the safety dogs. This type of rail has passed out of use, and today a wood rail is invariably used. These rails are generally made of maple and are usually $1\frac{3}{4}$ by $1\frac{3}{4}$ inches, being fastened to the guidepost by means of flat-headed wood screws. The guide strip is bored for the screws at intervals of 15 inches, and the holes are countersunk to let the heads of the screws lie well below the surface of the guide strip, this being necessary to prevent the guide shoes on the car from catching on a projecting screwhead.

*Kinds of Wood Used.* The use of a hard close-grained wood, such as maple, beech, birch, or cherry, for the guide rail is necessary to obtain durability, but the rail must not be too large, for the greater rubbing surface increases the friction, which in turn perceptibly increases the power required to operate the elevator. Moreover, the high cost of these hard woods and their liability to warp are further objections to making the guide and post in one piece. Still it is very essential that the guide be rigid in order to obtain steadiness and stability in operation and safety in case of breaking hoist ropes, thus throwing the safety dogs, in which event the guides cannot be too rigid.

For these reasons it has been found most desirable from every point of view to use a guidepost made of some less expensive wood—one which is not so liable to warp or get out of shape, such as well-seasoned pine or spruce. As a further precaution, the guidepost is usually made up of several thicknesses of 2-inch plank, three pieces 2 by 6 inches or 2 by 8 inches, surfaced on both sides and edges, and spiked together, forming a post of about 5 by $5\frac{1}{2}$ inches or 5 by $7\frac{1}{2}$ inches. Care is taken that one of the sides of each post, i.e., that side to which the guide is to be attached, is straight and true, and that, in setting them in place in the hatchway, they are set to plumb lines so

as to be parallel to one another and central between the back and front sides of the hatchway.

*Method of Attaching.* When these guideposts are in place the hardwood guides or rails on which the car travels are attached, thus giving the additional advantage of permitting the guide rails to be made in short lengths of about 4 feet. This permits the exercise of economy in the manufacture of the rails, as there is a lower percentage of waste when cutting short straight lengths. In order to insure perfect alignment at the joints, the ends of the guide strips are matched, that is, they are alternately tongued and grooved, the tongue on the end of one length fitting into the groove in the end of

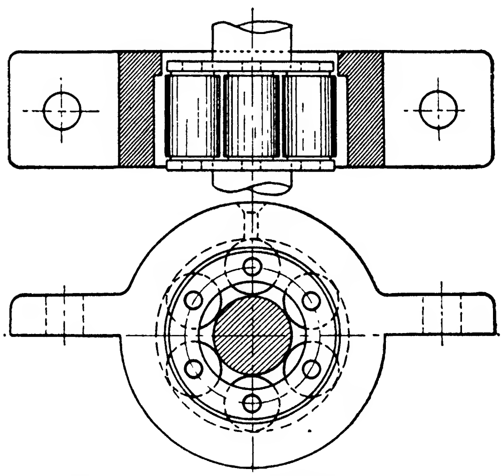


Fig 8. Details of Roller Bearings for Elevator Drum and Gears

the next. Holes for the wood screws which fasten them to the posts are also provided at about 3 inches from each end and at intervals of about 14 or 15 inches, as stated before.

**Introduction of Antifriction Roller Bearings.** The use of a traveling car or cage, of course, necessitated the use of a counterbalancing weight to offset the weight of the car, but this additional load on the journals and bearings of the drum and gear caused the machine to run stiffly. This required additional exertion on the part of the operator and thus detracted from the advantage gained by the convenience in handling goods. Attention, therefore, had to be given to this feature with the result that the "antifriction roller

bearing" was developed, comprising a set of rolls on pins running in circular frame a or cage made of two rings, as shown in Fig. 8. The rollers are usually made of cold-rolled Bessemer-steel shafting, $\frac{15}{16}$ inches in diameter, cut off in lengths of about $1\frac{1}{2}$ inches, chucked true in the lathe and bored for $\frac{5}{16}$-inch pins designed to run loosely, and faced on the ends. Six of them are then assembled between the two rings shown in Fig. 8, and a special box or case is provided for them to run in, as shown in Fig. 9. It will be noted that different forms of boxes are shown, the difference being only in the height of the center of the box above the bases provided for attaching them to the wood frame. This difference in height is due to the difference in the diameters of gears and pinions used, and to the need for keeping

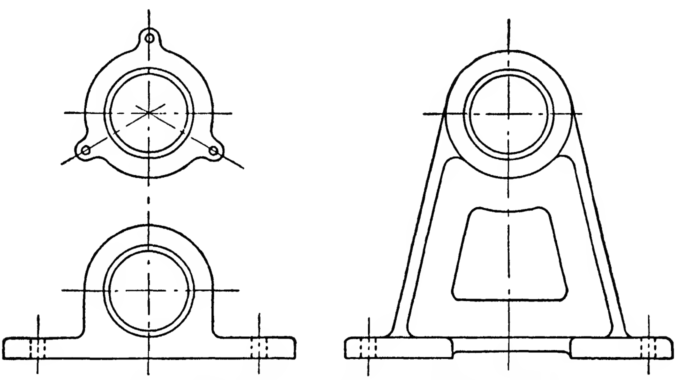


Fig. 9. Different Types of Boxes for Roller Bearings

the centers of the drum- and pinion-shafts within certain fixed limits determined by hatch size and rope-wheel, drum, and gear diameters. The difference is merely a matter of convenience, the box being always the same. The use of antifriction roller bearings in connection with hand-elevator gearing produces such a marked improvement in the ease with which the elevator can be operated that no experienced builder thinks of making a hand elevator without using them.

*Necessity for Oiling.* Although many makers claim that such bearings will run without oil, this is a fallacy; for, although they will run without lubrication, they wear out very quickly under such conditions, while, when properly lubricated, they will last for years. It is therefore both prudent and economical to lubricate them at intervals of once or twice a month with a small quantity of oil. Oil is

principally needed on the pins upon which the rollers run, and at the ends of the rollers where they bear against the shoulders of the journals and bearings. If oil is put into the box through the oil hole at the top, it will distribute itself properly, provided a sufficient quantity is used.

**Early Type of Counterbalancing Weights.** The first counterbalancing weights used were very crude affairs, being simply long rectangular pieces of cast iron with an eye cast in one end for attaching the rope from which they were hung. They were allowed to run in a pocket located against a wall at some distance from the hatchway, the weight rope being led from the drum and over a sheave set above the weight pocket, in a manner somewhat similar to the way in which the hoisting rope in the outrigger type of the old sling machine was led to the sheave outside the building.

*Means Employed to Keep Rope Plumb.* This arrangement was largely due to the room that several turns of the manila rope required on the drum, a circumstance which prevented leading the rope down inside the hatchway and locating the counterpoise weight alongside the guidepost. The same difficulty was met in the case of the hoisting rope because the room it required on the drum, when several turns of rope were wound upon it, made it impossible to keep the hoisting rope hanging plumb in the center of the shaft, thereby causing the cage to be pulled sideways against the guides and giving rise to considerable friction. To overcome this defect, it was customary to use a longer hoist rope, fastening both ends of the rope to the hoisting drum and passing the loop through a sheave about five inches in diameter in the lifting strap of the car, Fig. 7, the sheave being used primarily to avoid abrading the hoist rope. The car then hung in the bight of the rope, that is, the loop formed by attaching both ends to the drum as described, with the result that the drum wound up both portions of the rope, causing the car to be lifted centrally. In attaching the ends of the rope to the drum, space was left to permit the rope to be wound up, as shown in the illustration.

**Served-Rope Hoist.** The ropes were also kept plumb by using sheaves with V-grooves similar to that in the large rope wheel, the sheaves being firmly keyed on a shaft which displaced the log drum, and which also had the spur gear keyed on it. The sheaves were about 14 inches in diameter and the shaft about $2\frac{1}{4}$

inches, the gear remaining about the same as it had been on the drum machine, that is, about 24 or 26 inches in diameter with a

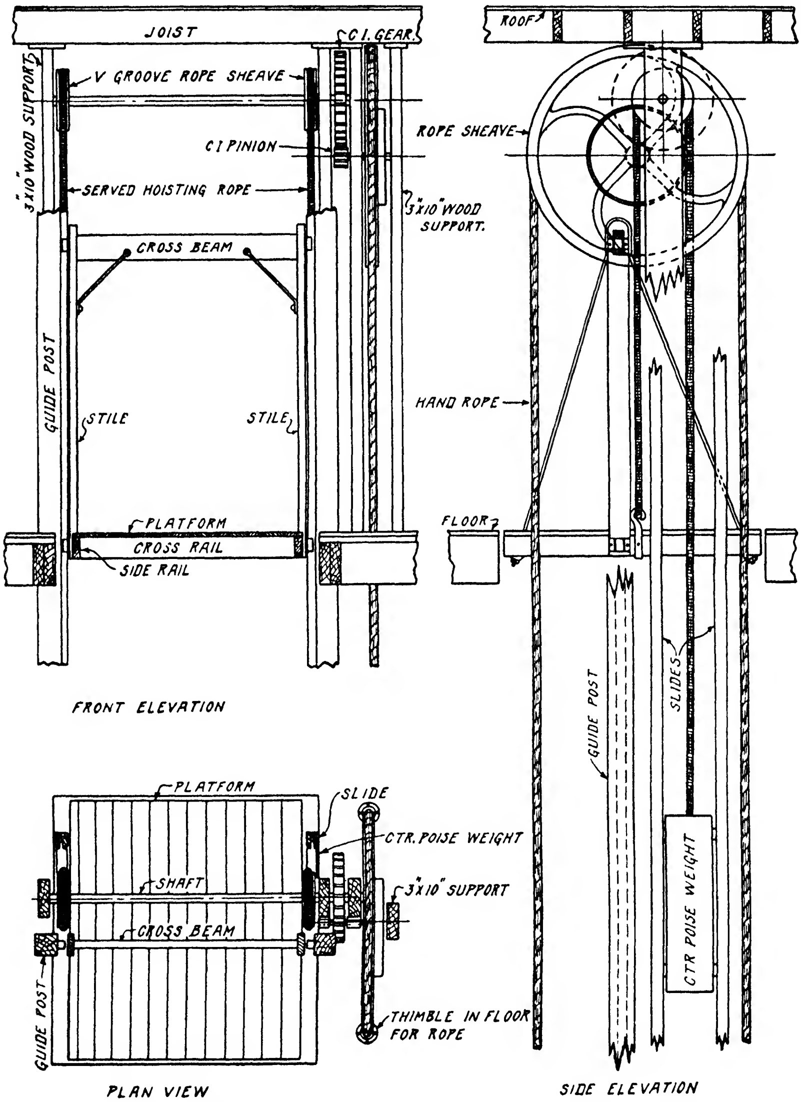



**Fig. 10. Details of Hand Elevator Showing Use of Served-Rope Hoist**

tooth pitch of 1 inch. Two hoist ropes about $1\frac{1}{4}$ to $1\frac{1}{2}$ inches in diameter were used, one end of each rope being attached to the bottom of the car close behind the guides, which, to allow of a central

lift, were set 4 inches to the front, as shown in Fig. 10. The other ends of the hoist ropes, after being led up to the top of the hatchway and over the 14-inch sheaves, were attached each to one of the counterpoise weights located on each side of the hatchway. With this arrangement there were two counterpoise weights and two hoist ropes, and the weights ran in open slides inside the hatchway. The load was lifted entirely by friction, that is, by the gripping of the hoist ropes in the V-scores of the 14-inch sheaves. To prevent wearing of the hoist ropes it was necessary to give the hoist ropes a protecting covering before putting them in place. This was done by "serving" or covering them with a thickness of marline wound on tightly while the ropes were stretched. When this protection wore through, it had to be renewed. The counterpoise weight on one end and the car at the other, as shown in Fig. 10, caused the ropes to sink into the V-grooves sufficiently to give the necessary traction to lift the load.

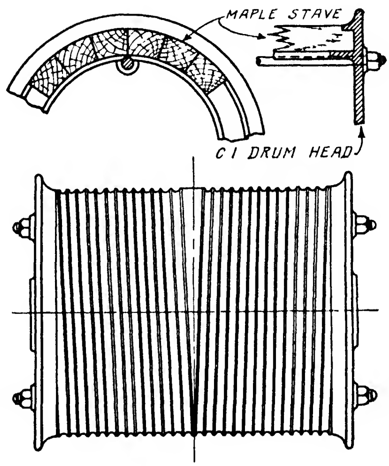


Fig 11 Details of Hand Elevator Drum

**Introduction of Wire Ropes.** When flexible wire ropes of circular cross section came into use they replaced the manila hoisting ropes on elevators, but the introduction of wire lifting ropes necessitated changing the drums so as to render them suitable for the purpose. As side-lifting manila ropes were already in use, the first wire ropes used were attached to the car and to the counterpoise weights in the same way. In order to give sufficient tractive force, the body of the drum used was made of wood, hard-maple staves, thoroughly dried, of keystone or wedge section being employed. These were fitted together as shown in Fig. 11, and held in position by two cast-iron flanges or drumheads, Figs. 11 and 12, bored and keywayed to fit the drum shaft. The resultant drum was put in the lathe, and the outside of the wood center, or barrel of the drum, was first trued or turned cylindrical; then a spiral groove to fit the cable was turned in the surface, the pitch of the spiral being a little

greater than the diameter of the cable or rope, in order to prevent the turns of rope on the drum from touching one another. When up in place, the wire ropes were wrapped around the drum about three times, one end of each cable being attached to one of the side rails of the car bottom and the other end to one of the weights. This arrangement gave sufficient traction to lift the required load, and the

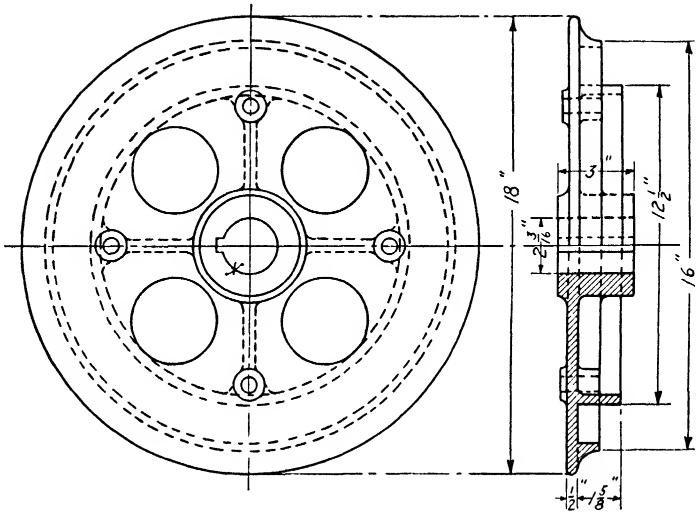



Fig. 12. End View and Part Section of Drum Heads for Wire-Rope Elevator

resultant machine was a more efficient, easier-running, and altogether more satisfactory elevator than any yet produced.

**Adoption of Center Lift.** The increasing height of buildings soon brought out defects which hitherto had not been suspected, for, with the increased travel of the car, it was found that the longer drums did not always wind the cables evenly. It was discovered also that the drums must be turned not only parallel but that both had to be of exactly the same diameter, a slight difference in their respective diameters causing a very perceptible difference in the level of the floor of the car during its travel in the higher buildings. As a result of the efforts to remedy this trouble, the first idea of lifting from the center of the crossbeam of the car, as shown in Fig. 13, was again brought into use. This proved entirely successful, and today the center lift is the popular form of hand-power elevator, although many makers for machines of short travel of one or at most two stories use the side lift; in these cases, however, economy in first cost,

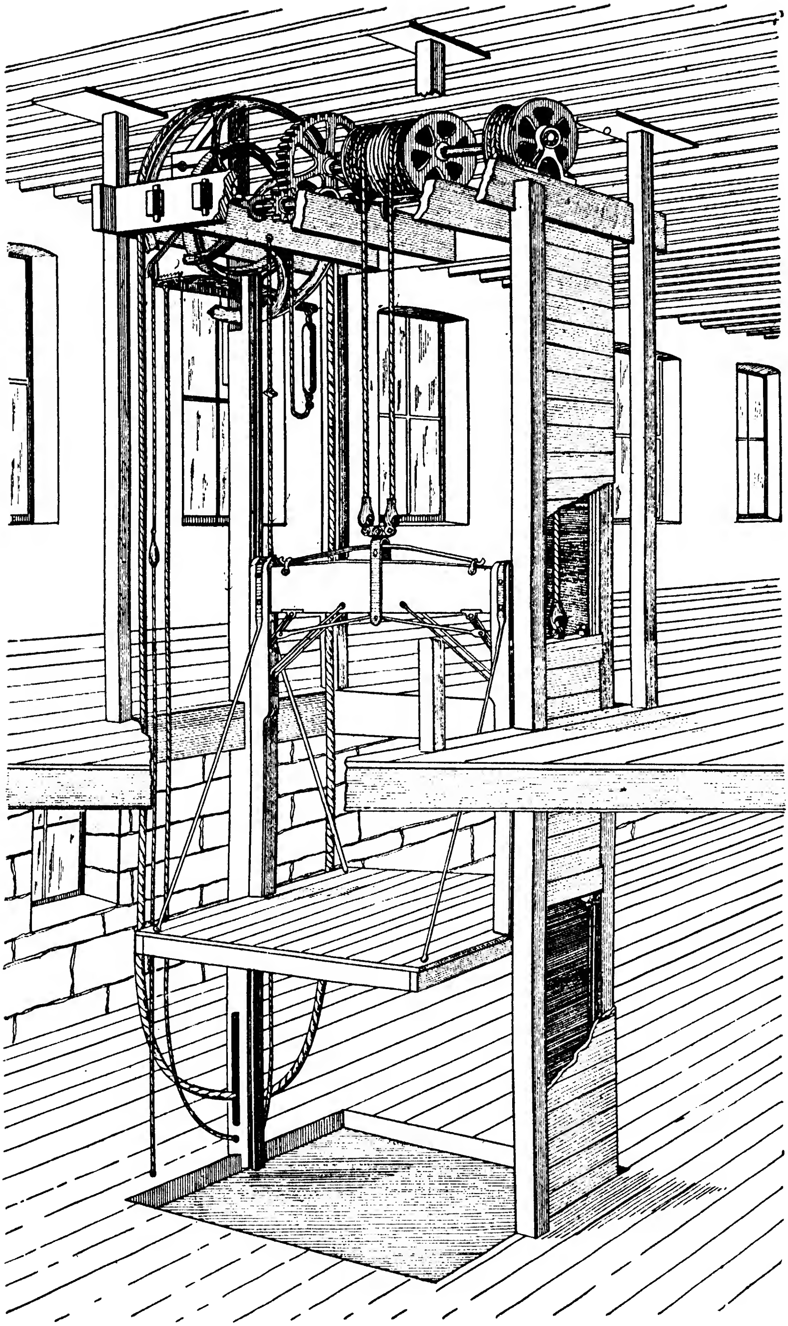


Fig. 13. Detail Views of Center-Lift Elevator with Grooved Drums and Wire Ropes

or the production of a cheap elevator, is the incentive. To further this object of low first cost, the drums are dispensed with and a V-grooved sheave is used, having the V form an angle of about 35 degrees. Wire rope instead of the s--ed manila rope is used, but such machines are neither desirable nor lasting, and they quickly destroy the lifting cables.

**Counterpoise Weights.** It was found to be desirable in most cases to make the counterpoise heavier than the car, so that when the car was empty and stationed at one of the lower floors, it would be possible to cause it to rise of itself to the upper floor at which it was wanted by simply releasing the brake, stopping it upon its arrival, and locking the brake. The improved counterpoise weight was formerly made in the form of a cast-iron frame having guides or guide shoes cast integral with the frame, the center of the weight being filled, or partly filled, with smaller weights, called sub-weights, until it was heavy enough.

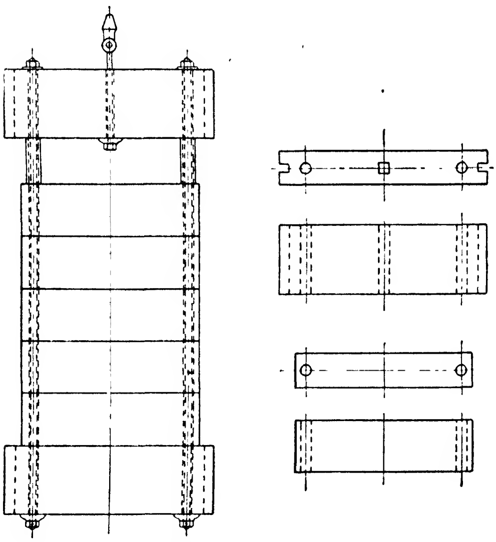


Fig 14. Elevator Counterpoise Weights, Showing Method of Fastening

*Modern Type.* The counterpoise weight of today is made up of a number of small weights, the top and bottom weights having the guide shoes cast on them, Fig. 14. The shoes travel on weight guides, which are similar to those on which the car travels but usually smaller in section, being either $1\frac{1}{4}$ by $1\frac{1}{4}$ inches or $1\frac{1}{2}$ by $1\frac{1}{2}$ inches. These weight guides are attached to guideposts similar to the car guides but smaller in section. They are usually made out of a single piece of timber 3 by 4 inches, 4 by 4 inches, or 4 by 6 inches, according to the size of the weight which is to run in them.

The sub-weights of the modern counterpoise weight are similar to the top and bottom weights, but have no guide shoes. Both the

main and sub-weights have holes cored through them near each end for the admission of long rods of $\frac{5}{8}$- or $\frac{3}{4}$-inch round iron. When the correct number of weights have been assembled, including the top and bottom weights, rods are passed through the holes, as shown in Fig. 14, the rods having been previously cut to the proper length and threaded for a distance of 5 or 6 inches at each end. Washers and double nuts are put on, and the whole is bolted together into one

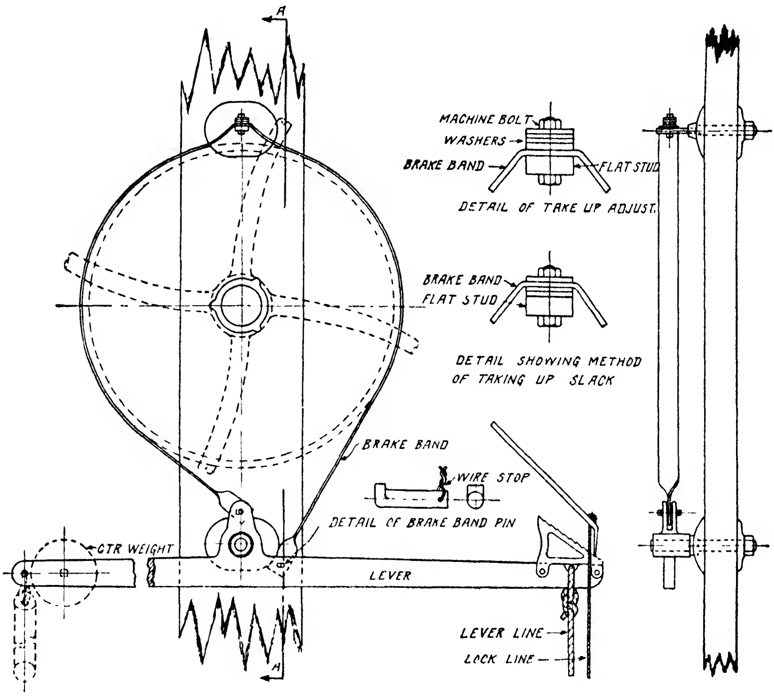



Fig. 15. Construction of Steel-Band Type of Elevator Brake

solid compact weight. The advantages of this method are obvious, as changes in the adjustments can easily be made at any time.

It was also found desirable in many cases to place the hand or pulley and brake ropes inside the hatchway, thus enabling the operator to ride up or down with the load on the car and be on hand to unload the goods. Here the extra amount of counterbalance weight proved to be of advantage, as the excess weight, being usually about 200 pounds, was also capable of lifting the man. The amount of overweight varies with the conditions of operation. If a truck is

constantly used with a load, the counterweight can be made to include also the weight of the truck, thus reducing to a minimum the manual labor required to operate the elevator.

**Brakes.** *Steel-Band Type.* In modern hand-power elevators the brake nearly always takes the same form, i.e., that of a steel band $\frac{1}{8}$ inch thick by $1\frac{3}{4}$ inches wide, Fig. 15, wrapped once around a pulley cast on the large rope wheel, the ends being attached to the

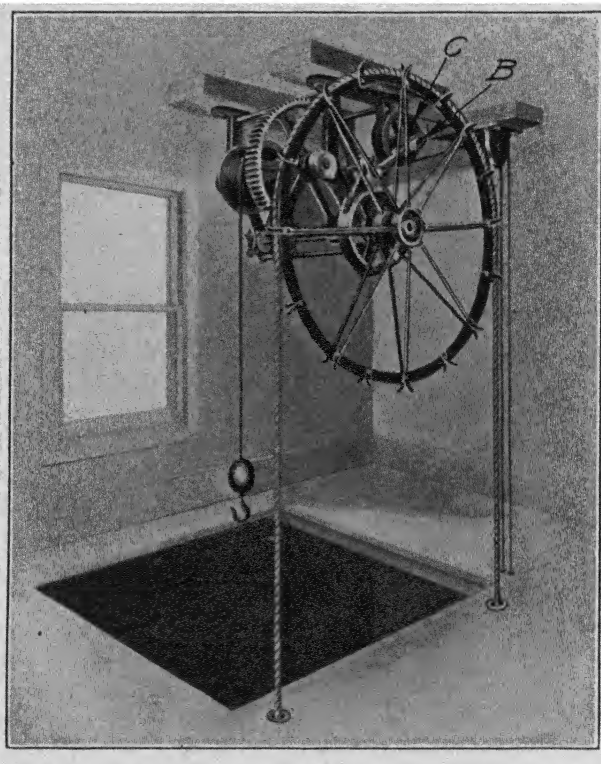



Fig. 16. Rope Hoist with Band Brake but without Ratchet and Locker
*Courtesy of Warsaw Elevator Company, Warsaw, New York*

brake lever and the center of the band supported by a stud attached to a suitable portion of the wood frame which carries the gearing. When the end of the lever is pulled down, it causes the band to clasp the pulley in a vise-like manner, and thereby stops the motion of the wheel and, through the medium of the intervening mechanism, the elevator also. A weight attached to the other end of the lever causes it to release the band when the brake rope is let free.

When it is desired to hold the platform at any one point after it has been stopped, the teeth of a dog or pawl called the locker are caused to engage in a ratchet bolted to the end of the brake lever, thus holding the brake on tight, so that the platform may be loaded or unloaded with safety. When it is desired to release the brake, a strong pull on the brake line causes the ratchet to become disengaged from the locker, and the brake is free again.

The brake shown in Fig. 16 includes the brake band and lever previously described, but minus the ratchet and locker. The end

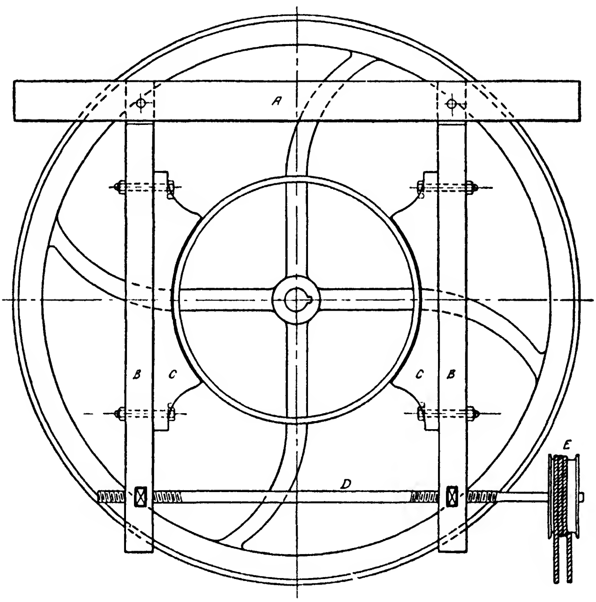



Fig 17. Construction of Wood Shoe Brake

of the lever is slotted, as shown at *B*, and in this slot a crankpin on the sheave *C* operates to move the lever. In the position in which the lever is shown the brake is on, but when the sheave is turned one-half revolution backward the brake is released. The machine here shown is a modified form of the old-fashioned sling machine previously described, but has a cast-iron drum instead of the wood drum, and a wire cable instead of the manila rope to lift the load. It is not a type to be recommended, as it is bad practice to use a cast-iron frame for a tensile strain, and it is only introduced to show this form of brake application.

*Wood-Block Type.* Another and very effective form of brake is shown in Fig. 17. It is somewhat crude in appearance, but it is found in practice to be very easy of operation and very powerful; it also has the advantage of not having to be locked, as it stays in whatever position it is placed. Its chief disadvantage is the necessity for using both hands when lowering a load, one hand being used for applying and the other for releasing the brake as occasion may require.

Referring to the illustration of this brake, *A* is an upper or crossbeam made of 3- by 4-inch hard wood, into which are tenoned two vertical pieces *B* of the same dimensions. These tenons *B*, are made to fit somewhat loosely into their respective mortises and are held in place by bolts or pins. To them are bolted two brake shoes *C*, usually of maple, sawed out in the band saw to fit the brake pulley. The vertical parts of this yoke are connected at their bottom ends by means of a long rod *D* of $1\frac{1}{4}$-inch round iron having at the proper distance apart a right- and a left-hand screw. This rod passes through the lower ends of the vertical pieces *B*, nuts fitting these screws being let into mortises in the pieces of wood. A sheave *E* is keyed on one end of the rod, and a rope attached to the periphery of the sheave is wrapped several times around it, the ends being led down the hatchway. Pulling on one side of the line turns the rod in a direction that causes the right- and left-hand screws and nuts to clamp the pulley in a tight grip; while pulling on the other side of the line, of course, reverses the motion and releases the brake.

**Safety Dogs.** *Unreliability with Side-Lift Machine.* With the side-lift machine suitable safety devices had been difficult. If only one cable broke, the immediate result was to rack the car, for it is assumed that the hoisting cables would break only when under strain or, in other words, when the car was loaded and the sudden letting down of one side of the loaded platform would distort the frame, throw it out of square, and tend to force the guideposts apart, thus ruining a most important member of the apparatus. Consequently the efforts of the makers were directed to devise a scheme to throw the safety dogs on both sides simultaneously with the breaking of either of the cables. To accomplish this end, both counterpoise weights were connected with the dog-throwing device by means of idle lines. These lines were not expected to do anything unless a

cable broke, when the weight in dropping would pull the idle line, or safety line, as it was called, and it in turn would throw in the dogs.

Although this looks like a very simple and ingenious arrangement, it proved in practice to be quite the contrary. The safety lines being of manila, frequently became tangled with other moving parts of the machine; and sometimes persons operating the elevator, being ignorant of their purpose and regarding them as superfluous, would cut them loose and remove them entirely.

*Relative Simplicity with Center-Lift Type.* With the adoption of the center-lift form of hand elevator the problem of the safety device became simplified, because where the lift is from one point, and that point the center of the car, it was only necessary to make the lifting strap movable through a few inches in a vertical line and arrange it so that tension on the lifting cables would hold it up to its highest point of travel. Operation could then be obtained by the application of a strong spring to throw the strap down when the tension on the cables was released. This strap was connected with the safety dogs or nippers by means of suitable levers, and the object sought was accomplished.

**Floor Thimbles and Rope Guides.** When the hand rope runs through the floors of the building, a cast-iron thimble is inserted in every hole in each floor through which these ropes run. Thimbles for large ropes have a hole about 3 inches in diameter and a flange about $1\frac{1}{2}$ inches in width through which are drilled holes for wood screws to hold them firmly in the floor. The thimbles for the smaller ropes have a hole 1 inch in diameter and a similarly proportioned flange. When the hand ropes run inside the hatchway, guides are attached to the trimmers of the hatch, but only at the top and next to the lower floors. The rope guides are also of cast iron, shaped as shown in Fig. 18, and are attached by lag screws to the hatch trimmers, as shown on Fig. 19. These guides are necessary to keep the hand rope from swinging to and fro, when the elevator is in operation. The

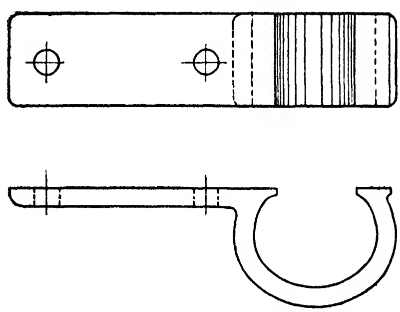


Fig. 18. Cast-Iron Rope Guide

pulling on the rope during the process of hoisting and the sudden application of the brake in lowering produce a violent swaying, pendulum-like motion in the rope when these rope guides are not used, thereby causing the rope to become wedged in between the platform and hatch trimmer, and frequently making the rope jump from its place in the rim of the large rope wheel. However, where these rope guides are used this trouble is eliminated.

**Importance of Fundamental Principles.** It may seem to the student who reads this article on hand elevators for the first time that the description of the historical development of these improvements is unnecessarily diffuse, but it is suggested that the reader withhold his judgment until he is better acquainted with the subject of elevators. He should take pains at this time to impress on his mind all the salient features of this particular type of machine, as well as the causes which led to its development. It is highly essential that he should do this for the reason that he will find these features constantly recurring in the other types of elevators described later. He will find that the principles evolved in the development of this type of elevator have been introduced in a multitude of cases in the more advanced types of elevators, for the simple reason that nothing better has so far been devised. The experience gained in the use of the scored drum; in the method of scoring it; in the use, construction, and application of the counterpoise weight; in the use of guides, safeties, and sheaves; in the methods of directing the cables, of braking, of stopping, and of locking; in the use of traction by using several turns of cable around drums, and in many other ways, has been of great service to the elevator builder in the successful development of the modern more powerful and rapid elevators which depend on other

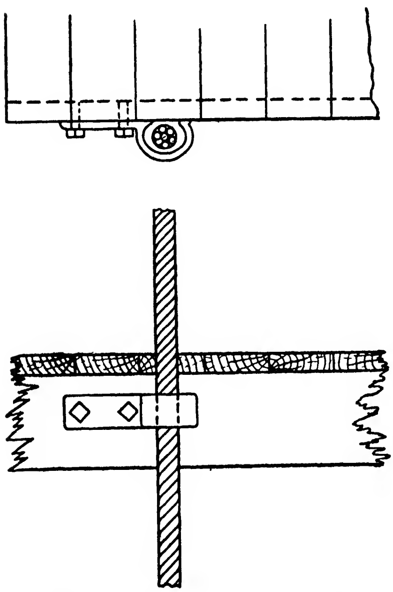


Fig. 19. Method of Fastening Rope Guide to Hatch Trimmer

sources of power for their operation.

### SPECIAL TYPES

In the preceding pages the general construction and development of the hand-power elevator have been briefly described. Many modifications of this machine are found at times to be desirable in order to suit varying conditions, and two or three of the most important variations will now be discussed.

**Lift with Pulling Rope Near Weight Slide.** Sometimes it is found desirable to locate the hand or pulling rope at a side of the hatchway adjacent to that on which the counterpoise weight is to be placed. In such case the hoisting drum is not arranged on the same shaft with the weight drum, but each drum has a separate shaft with appropriate journals and boxes. In making up the frame which carries the gearing, the weight drum is located directly behind the hoisting drum, as shown in the plan view,

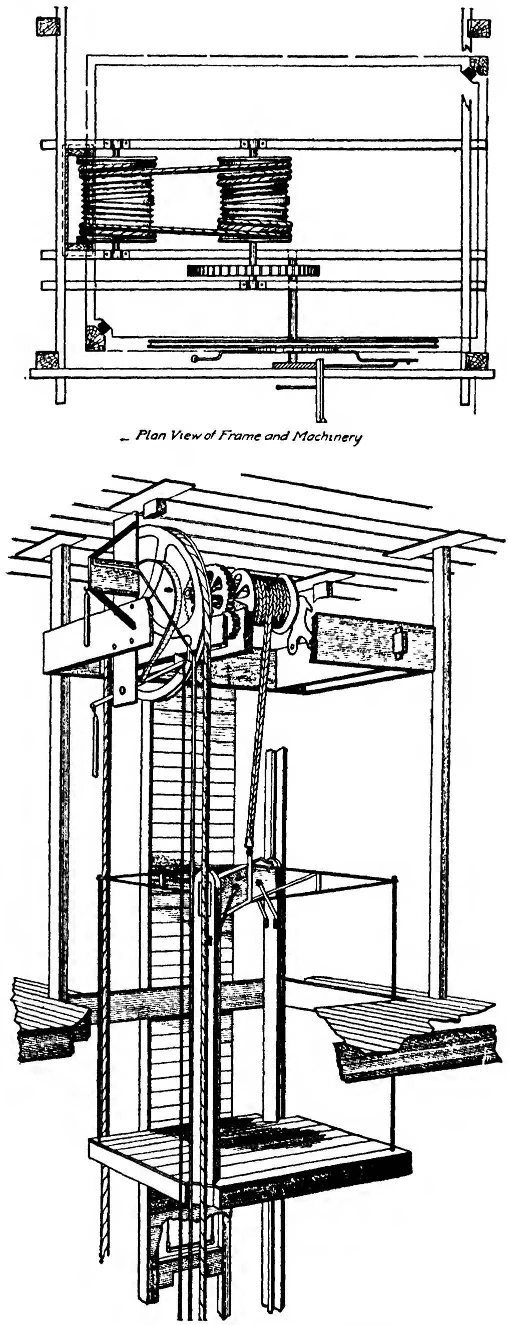



Fig. 20. Details of Corner-Post Elevator

Fig. 20. In that case there are only two cables, and each cable runs directly from the car up to and twice around the hoisting drum, thence across to and once around the weight drum, and from there down to the weight, the general arrangement in other respects being the same.

**Corner-Post Elevator.** In cases where the entrance to the platform from all sides is desired, it is found essential to

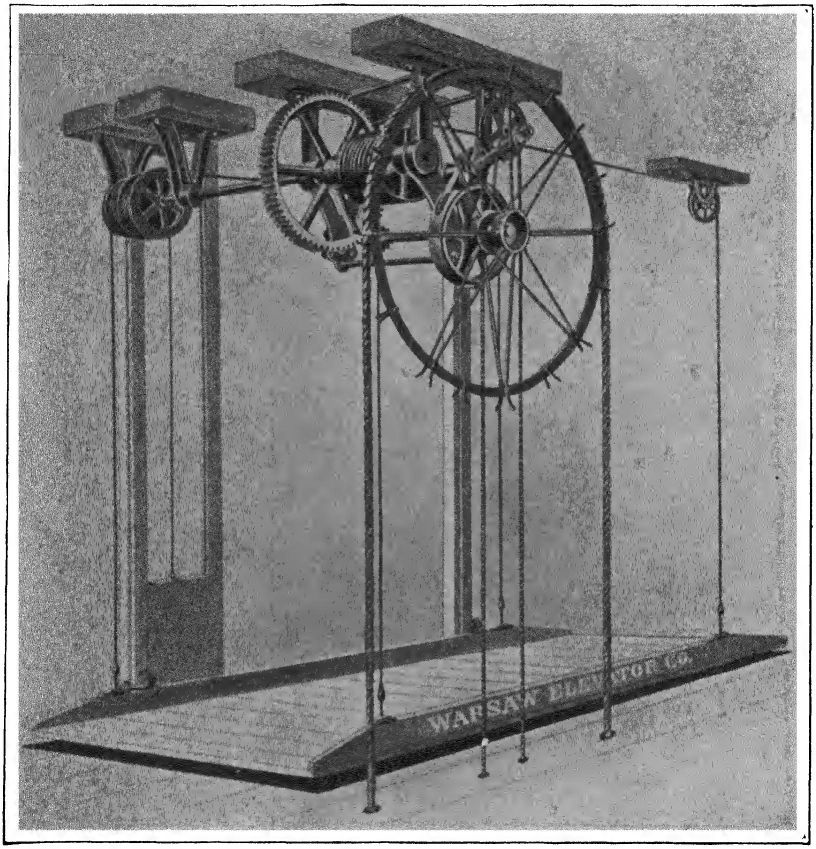


Fig. 21. Carriage Elevator with Two Hoisting Drums and Guides Only on One Side
*Courtesy of Warsaw Elevator Company, Warsaw, New York*

place the guides at opposite corners of the hatchway, forming what is called a corner-post platform. This feature is combined in the elevator shown in Fig. 20.

**Wagon and Carriage Lift.** Where it is desired to lift long and bulky articles, such, for instance, as wagons and carriages, another and different arrangement is resorted to, as shown in Fig. 21. In this

case two drums on one shaft are used, both being hoisting drums, but one being made longer than the other for the purpose of winding and unwinding the weight cable. Two hoisting cables are used on the hoisting portion of each drum.

The frame which carries the overhead work or gearing is set above one end of the long platform, and two hoisting cables are led directly down from these drums to one end of the two sides of the platform. To lift and lower the other end in unison, two additional cables are attached to the same drums and led from there to sheaves located at the opposite end of the hatchway, and thence down to the other end of the platform. By this arrangement the platform is lifted

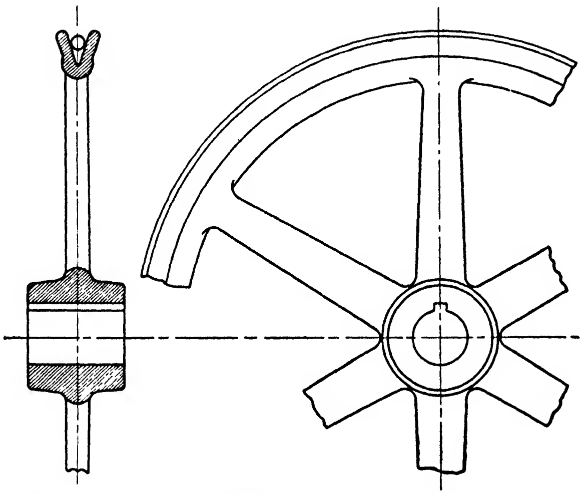


Fig. 22. V-Grooved Hoisting Sheave for Light Hand-Power Elevators

at its four corners, so that no stiles or uprights are needed, nor is a crossbeam used, and guides are required on only one side of the platform. It is customary to utilize the weight slide as one of these guides, a separate guide being placed near the other end of the platform on the same side as the weight slide. The arrangement and construction have been shown in detail in previous illustrations.

**Light-Load Sheave-Type Elevators.** When the load to be lifted does not exceed 800 pounds another modification is used. Every part of the framing and gearing is made the same but somewhat lighter in proportion, the principal difference being in the use of a sheave instead of a hoisting drum. This sheave has a V-shaped groove for the cable to run in, one end of the cable being attached

to the top of the car or platform. After being carried up and over the sheave, the other end of the cable is attached to the weight, the sheave being made large enough in diameter to span the distance between the center of the car to the center of the weight slide, so that both

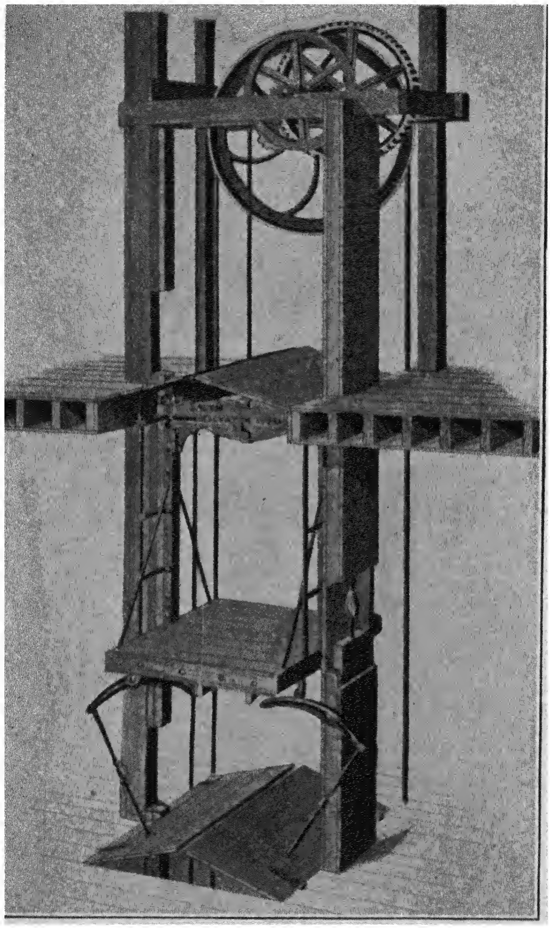


Fig. 23 Hand-Power Elevator with Automatic Safety Doors
*Courtesy of Salem Elevator Works, Salem, Massachusetts*

ends of the cable hang plumb. Fig. 22 shows the type of sheave used, and Fig. 23 its application to an elevator with the added feature of automatically operated trapdoors.

Before closing this article on lifting machines operated by hand power, it might be well to give a brief description of two other forms

of the machine which, although not so frequently met with as those just described, are nevertheless very useful in their way and of interest on that account. We refer to the basement or sidewalk lift and the dumb-waiter.

## BASEMENT ELEVATOR

This machine, as its name implies, is used for lifting loads through the height of only one story, usually from the level of the basement floor to that of the ground floor, or of the sidewalk.

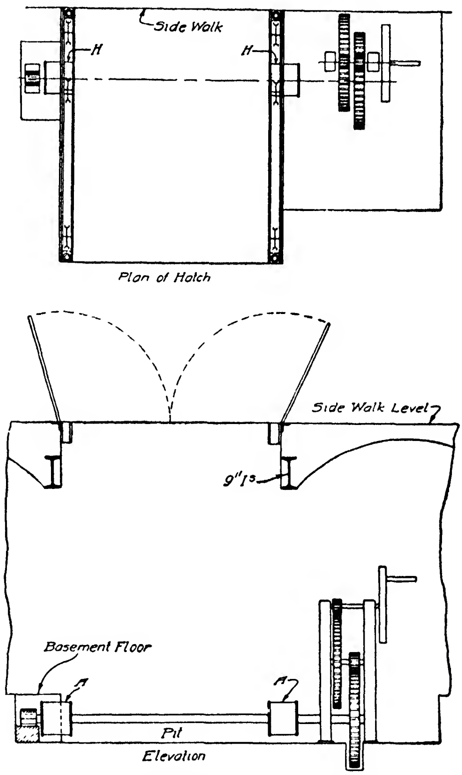



Fig 24. Gear Drum and Sheave Arrangement for Basement Elevator

**Four-Chain Type.** *General Construction.* Figs. 24 and 25 give a good idea of the general form of the 4-chain type. The platform on which the load is placed is lifted at the four corners by means of chains which run over sheaves or pulleys, their bearings being set in frames which bring them level with the ground floor or sidewalk. The lifting chains, after passing over the sheaves, are led down to the hoisting drums *A* below, Fig 24. The drums are keyed securely on a shaft, to one end of which is attached a spur gear revolving in bearings located below the level of the basement floor. The drums are placed a sufficient distance apart on the shaft to allow the platform to pass between them, and far enough below the floor level to permit the platform to descend to the level of the floor without striking the shaft.

The spur gear, previously mentioned as being at one end of the shaft, meshes with a pinion which is a portion of a train of gears and pinions keyed to shafts running in bearings in a cast-iron frame.

The first or upper shaft is provided with a hand crank, Fig. 24. The second shaft in the winch has a brake pulley keyed to it, a brake band and lever being provided to effect the application of the brake

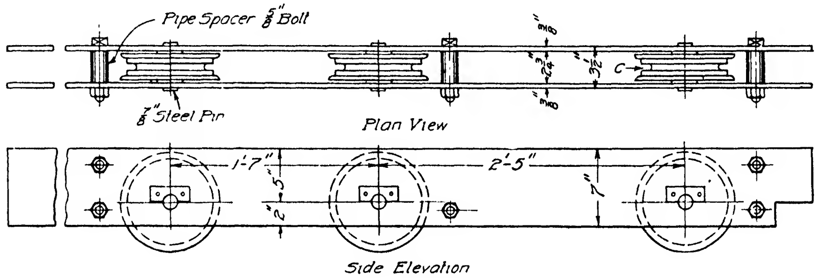


Fig 25 Plan and Side Elevation of Sheave Bars for Basement Elevators

when needed. There is also a pawl or dog attached to the frame which, when called into play, drops or meshes into the teeth of the first gear. A purchase of about 26 or 30 to 1 is usually employed.

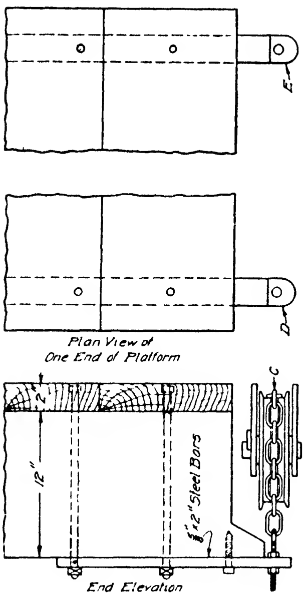


Fig. 26 Details of Platform for Basement Elevators

*Sheaves and Sheave Bars.* Fig. 25 shows the construction, details, and typical dimensions of a set of sheave bars. They are usually made of bar iron $\frac{3}{8}$ to $\frac{1}{2}$ inch in thickness and either 6 or 7 inches wide, depending on the length of bar and load to be lifted. The sheaves are about 9 inches in diameter, and are of cast iron bored to receive a $\frac{7}{8}$-inch steel pin which is kept in position by means of a flat plate at one end. This plate, as shown in Fig. 25, is bolted to one of the bars by means of tap bolts and fits into a slot cut halfway through the pin near one end. This arrangement not only keeps the pin in place but effectually prevents it from turning, thereby making it necessary for the sheave to revolve on the pin. Oiling facilities are provided in the sheave in the form of a small oil cup screwed into the hub of same.

*Chain Arrangement.* The sheaves are provided with a double groove, which permits the chain to lie evenly in the sheave, as shown

at $C$ in Figs. 25 and 26. The sheaves, as can be seen from the side elevation in Fig. 25, are set in position in the bars so that the flanges of the sheaves are slightly below the upper edge of the bars, and so that the two outer sheaves on each bar permit the chains to drop plumb to the irons $D$ and $E$, Fig. 26, on the lower edge of the cross rail of the platform to which they are attached. The intermediate

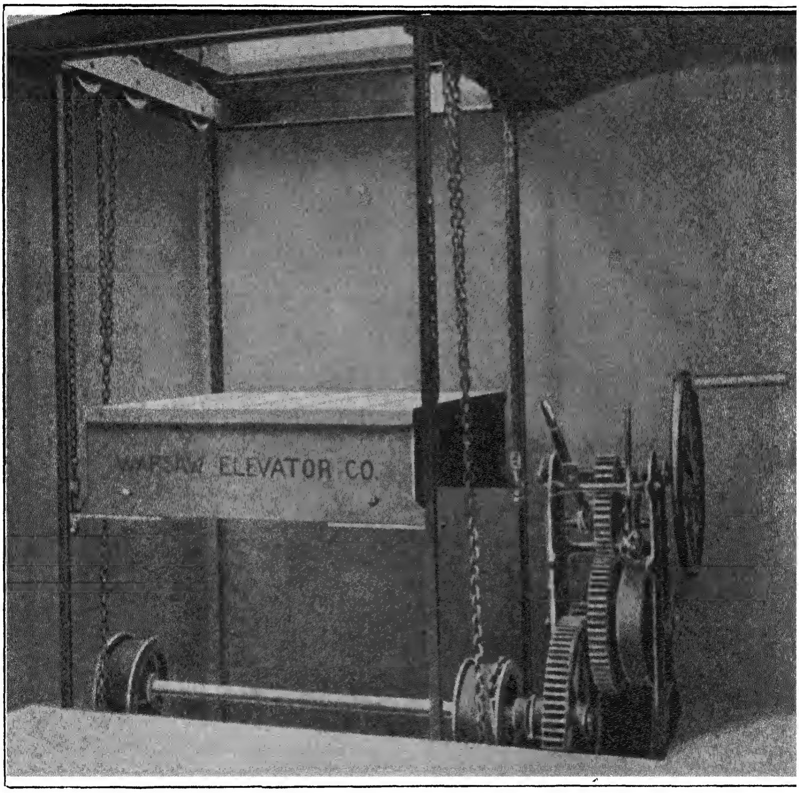



Fig. 27. Complete Hand-Power Basement Elevator
*Courtesy of Warsaw Elevator Company, Warsaw, New York*

sheave, which is only a leader for one of the chains, is usually set about 18 or 19 inches from one of the end sheaves. The drum shaft is set in the pit, so that the hoisting sides (see $H$, Fig. 24) of its drums are vertically below the center point between each of the two pairs of these nearby sheaves. The end of each chain is attached to each drum by a bolt passing through the last link, and through a hole drilled in the rim of the drum and secured there by means of a nut.

*Platform.* The platform is made of 2-inch oak, maple, or hard pine. Fig. 26 shows a top and end view of one side of the platform. The cross rails are about 12 inches deep, and the pit or depression in the floor is about 14 inches deep, thus permitting the top of the floor of the car to come even with the floor of the basement. The arrangement and mounting of the platform, chains, and sheaves can be seen in Fig. 27.

*Guides.* The platform travels on guides, usually located one at each corner. These guides are made in several forms, being usually either of 2-inch iron pipe or of 3- by 3- by $\frac{3}{8}$-inch steel angles, or in some cases of square bars of iron $1\frac{1}{4}$ by $1\frac{1}{4}$ inches. All are satisfactory if sufficiently rigid. The upper ends of these guides are fastened to the frames which carry the chain sheaves, and the lower ends are fastened below the floor of the pit in which the shaft and gearing of the machine for this type of elevator are located.

When pipes are used as guides and supports as well, flanges are screwed on the lower ends of the pipes of sufficient outside diameter to give them a firm bearing when bolted to the pit floor. As the upper ends of the guides terminate just below the sheave bars, the cross rails of the platform must be deep enough to keep the guide shoes on the rails when the platform is level with the first floor or sidewalk.

When either pipe or angle irons are used for guides they frequently form supports for the sheave frames, but where possible the ends of the sheave bars are made to project beyond the guides a sufficient distance to permit them to rest on ledges left at the ground level to carry them.

*Operation.* Raising the platform is accomplished by simply turning the crank generally in a clockwise direction, Fig. 27. A pawl or dog attached to the frame of the winch directly above the first spur gear drops into and engages the teeth of the gear, thereby preventing its turning in the opposite direction. The car is hence locked against downward movement during the period of raising and of loading at the upper level.

To lower the car it is necessary to first raise it slightly by turning the crank in the proper direction sufficiently to relieve the pawl. While holding it in this position seize the brake lever with one hand and apply the brake. Then, holding the brake firmly, remove the

hand from the crank and with this hand throw the pawl out of the way of the gear teeth. Afterwards slide the crankshaft lengthwise, so as to disengage the pinion from the gear teeth and use the brake for lowering. After the car is stopped at the lower level, the pawl is again thrown against the gear. Before hoisting again, the crankshaft must be moved lengthwise until the pinion engages with the gear. The reason for throwing the gear and pinion out of mesh when lowering is to remove the danger of accident inherent in the rapidly revolving crank. Slipping the pinion and gear out of mesh allows the crank to remain at rest while lowering. A lock or clutch on the frame of the winch may be used to keep the crankshaft in the proper position during raising or lowering.

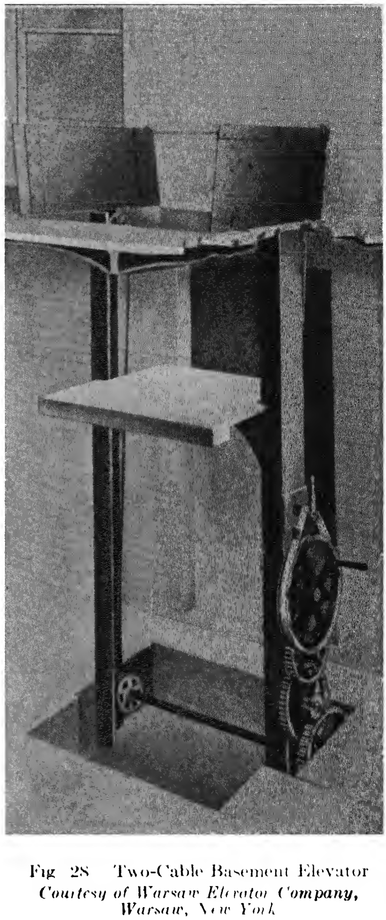
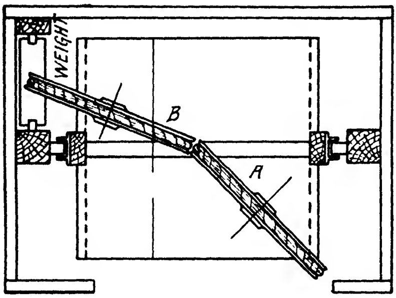


Fig. 28. Two-Cable Basement Elevator
*Courtesy of Warsaw Elevator Company, Warsaw, New York*

The pawl must not be thrown into gear while the platform is moving downward, for under such conditions it will break the gear teeth. The form and arrangement of the crankshaft, collar, and gear-shaft lock will be clearly understood by reference to Fig. 27.

**Two-Cable Type.** A simple form of the basement elevator or sidewalk lift, suitable only for light loads, is shown in Fig. 28. It includes a train of gearing and a brake similar to that previously described, but these are all supported on studs located near the bottom of the 6-inch side of a 4- by 6- by $\frac{1}{2}$-inch angle iron, the 4-inch side of which serves as one of the guides for the platform. The other guide consists of a similar angle on the opposite side of car. The platform is lifted by means of two $\frac{1}{2}$-inch wire ropes instead of four chains. These wire ropes are




attached at one end to the hoisting drums, which are keyed on a shaft about 2 to $2\frac{3}{16}$ inches in diameter, and are carried up the runway and over two sheaves running on studs attached to the upper ends of the same angle irons, and from there down to the car or platform to which they are attached. The operation of the machine is similar to that of the kind first described, but the entire apparatus is simpler in construction and more easily installed, although not so powerful.

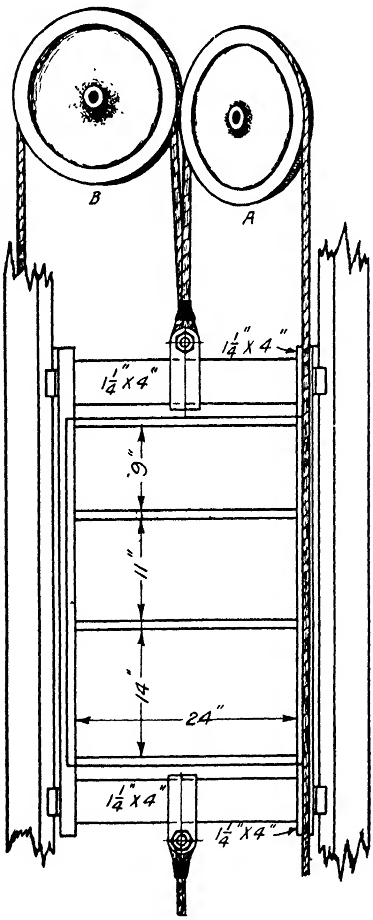



Fig. 29. Details of Domestic Dumb-Waiter

### DUMB-WAITER

The dumb-waiter is chiefly used in dwellings, restaurants, and public institutions for the conveyance of food from the kitchen to the dining room, and in stores for sending light packages from one story to another. The loads conveyed vary up to 100 pounds and in exceptional cases up to 150 pounds.

In dwellings the load seldom exceeds 20 to 25 pounds; in restaurants the load may be as great as 40 pounds; in public institutions, such as asylums, hospitals, etc., the loads are often as great as 90 pounds; and in stores the capacity required sometimes reaches 125 pounds.

The machine comprises a box with shelves so built as to have

one or two open sides, the number depending on the layout of the building. This box travels on hardwood guides extending the entire length of the run, or travel, of the dumb-waiter. These guides are usually one inch square but vary in types of different make. The mechanism used for their operation differs somewhat according to the service they are to perform and the ideas of the manufacturer.

**Domestic Type.** Fig. 29 shows the general construction and dimensions of a well-constructed and popular type suitable only for light domestic service.

*Box Construction.* The box may be made of $\frac{7}{8}$- or $1\frac{1}{8}$-inch pine, fir, or hardwood, mounted on a $1\frac{1}{4}$- by 4-inch carrying frame consisting of two uprights to which the four guide shoes are attached, and an upper and lower crossbeam, the ends of which are mortised and pinned to the uprights.

*Tackle.* To the center of the upper beam a $\frac{3}{4}$-inch manila rope is attached by passing it around the crossbeam and either splicing it or seizing it. This rope is carried to the top of the run and passed over a sheave $A$, hung in a diagonal position, so as to lead the rope down near the right-hand corner of the box, as shown in the plan view. It is then led down and passed around another sheave set in a similar position near the bottom of the run, and thence up to the lower beam of the box, where it is fastened. A $\frac{1}{2}$-inch manila rope is also attached to the upper beam of the box and is led over the sheave $B$ and attached to the counterbalance weight, the position of which may be seen in Fig. 29. This weight, which must not be heavier than the box, runs on hardwood guides about $\frac{3}{4}$ inch square. The $\frac{3}{4}$-inch rope which lifts the box is intended to be used also as a hand rope for operating it, and it is for this reason that it is led down near one of the front corners of the box. It can be readily seen that by pulling the rope in one direction the box will be caused to move in the opposite direction. With such an arrangement a force equal to the load to be lifted plus the friction must be exerted, and, in order to reduce the friction to a minimum, the wood or cast-iron sheaves are often mounted on patent bushings, which are simply small roller bearings.

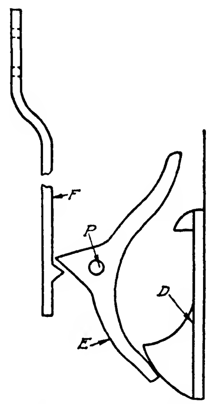


Fig. 30. Box Locking Device for Dumb-Waiter

*Box-Locking Device.* Fig. 30 shows the box-locking device employed. It comprises a cast-iron cam *D* screwed firmly to the box

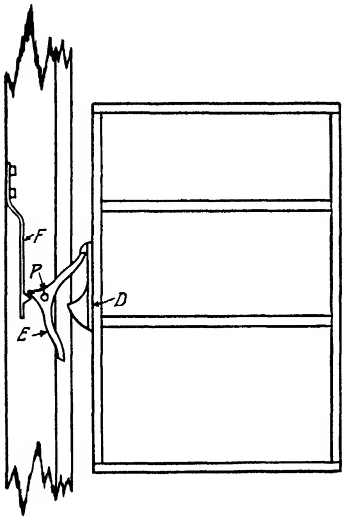



Fig. 31. Dumb-Waiter Locked in Position

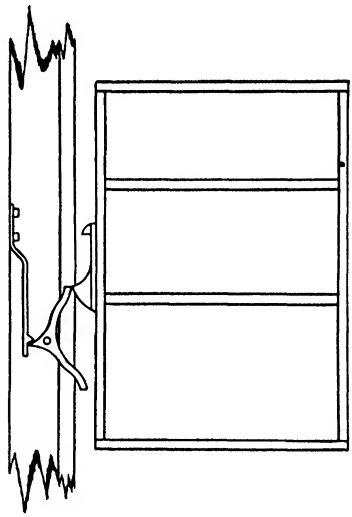


Fig. 32. Position of Lock on Slightly Raising Car

and a cast-iron latch *E* attached in the runway which vibrates on a pin. *F* is a steel spring for holding the latch firmly in position when the box is brought to rest from either direction of movement. A separate latch and spring are required for every doorway in the run.

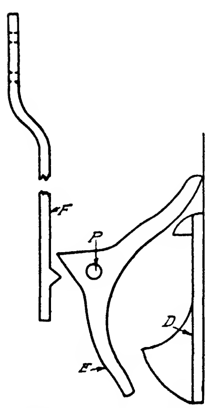



Fig. 33. Cam Position with Car Going Up Just Before Locking

When the latch and cam are in the position shown in Fig. 31 the box cannot descend. To lower the box it is first slightly raised by pulling down on the hand rope until in the position shown in Fig. 32, after which it can be lowered without obstruction.

When a loaded box is to be hoisted to any particular doorway to be unloaded, the box is pulled up until the cam *D* is in the position shown in Fig. 33 and, on being raised an inch or two higher, the top part of the latch will engage with the upper part of the cam, as shown in Fig. 31, and the box will then stand where left. A study of these sketches show that it is impossible, when the parts of this appliance

are properly proportioned, for the box to be accidentally stopped while moving in either direction.

**Medium-Load Type.** Fig. 34 shows the same dumb-waiter arranged with a double purchase to enable the operator to lift heavier loads. The general arrangement is similar to that in Fig. 29, except that extra sheaves are attached to the upper and lower beams of the box by means of wrought-iron straps $C$ and $C_1$. These straps are made of 2- by $\frac{3}{16}$-inch strap iron bent to fit the beams to which they are fastened by bolts. The sheaves are slipped between the prongs of the forks thus formed and run upon pins fitted into the ends of the straps. The lifting rope is passed around each sheave and the ends are attached to some fixed points in the elevator runway. The effect of this arrangement is to cause the box to travel only one-half the distance the hand rope is pulled, but it gives the operator double lifting power. A car-locking device of the nature previously described is employed.

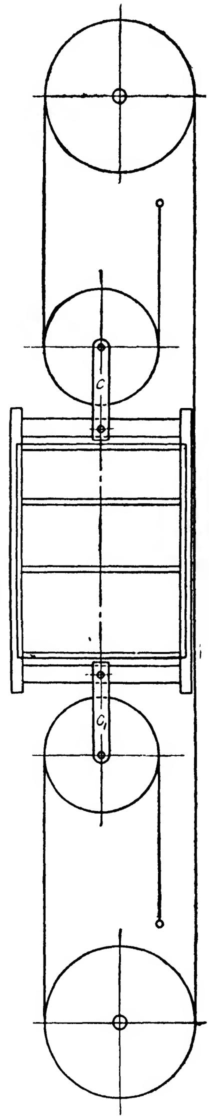


Fig. 34. Details of Medium-Load Dumb-Waiter with Extra Sheaves

**Heavy-Service Type.** Fig. 35 shows another form of dumb-waiter which is used for lifting heavier loads, such as are common in stores, public institutions, etc. It employs two sheaves of different diameters mounted on a shaft running in bearings. The upper portion of the figure gives two views of the sheaves and shaft. It will be noticed that the purchase is obtained by using sheaves of different diameters. The usual dumb-waiter box and counterpoise weight running on their respective guides are employed. Two separate ropes are used, one being an endless hand or operating rope, which runs in guides the same as the hand rope for a hand-power elevator, and the other being a lifting rope, one end of which is attached to the box and the other to the counterpoise weight.

When the hoisting sheave is not large enough to span the dis-

tance from the center of the runway to the plane of movement of the counterpoise weight, it is usual to use two idle sheaves to deflect the lifting rope, these being arranged as shown in Fig. 36, so that the lifting rope is in contact with half the circumference of the lifting sheave, which is necessary in order to obtain the required amount of friction.

*Sheave-Locking Device.* The most interesting feature of the heavy-service machine is that it will stand wherever it is left, either

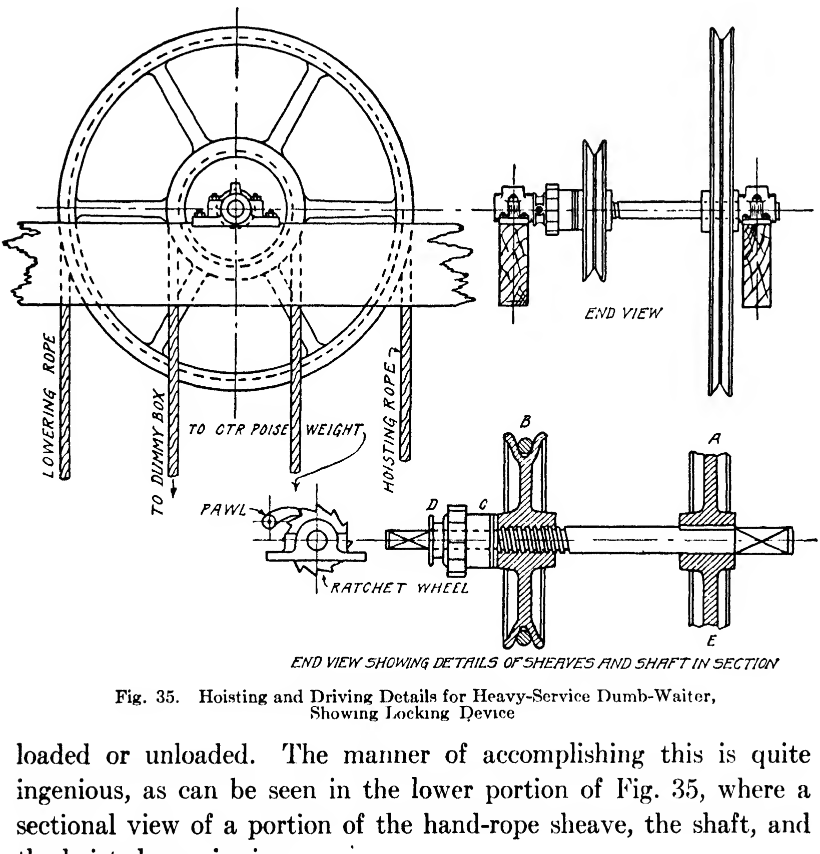



Fig. 35. Hoisting and Driving Details for Heavy-Service Dumb-Waiter, Showing Locking Device

loaded or unloaded. The manner of accomplishing this is quite ingenious, as can be seen in the lower portion of Fig. 35, where a sectional view of a portion of the hand-rope sheave, the shaft, and the hoist sheave is given.

By reference to this figure it will be seen that the hand-rope sheave *A* is keyed solidly on the shaft, but that the inside surface of the hub of the hoist sheave is threaded, and that a portion of the shaft is threaded to fit the hub of the sheave, an easy fit being made, so that the sheave will never stick or jam.

On the plain portion of the shaft beyond the thread is fitted a collar *C*, which is free to revolve on the shaft. This collar has on its periphery a series of saw-like ratchet teeth. When the shaft is in place in its bearings a pawl or dog attached to the framework falls by gravity into these teeth, thus permitting this loose collar to revolve only in one direction. Beyond this loose collar is fitted another collar *D*, which is held rigidly in position by either one or more set screws or by a pin passing through the shaft and collar.

Its manner of operation is as follows: The shaft being in its place in the frame and a load being placed in the box, which is hung from the rope at *B*, the tendency of this load will be to jam the sheave up tight against *C*, which, in turn being kept from revolving in that

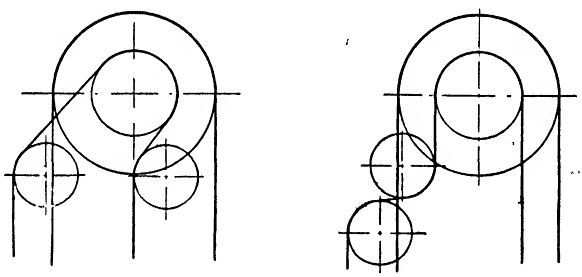


Fig. 36. Diagram Showing the Use of Idle Sheaves to Control Position of Lifting Rope

direction by the pawl engaging in the ratchet teeth on its rim, remains stationary.

In order to hoist the load, it will be necessary to pull down on the *E* side of the hand rope. This action also screws the hoist sheave tight against the collar *C*, and it in turn against *D*, but the action of the pawl on the ratchet permits the revolution of collar *C* in the direction for hoisting and, whenever the action on the hand rope stops, the box becomes and remains stationary.

Lowering the load is done by pulling down on the *A* side of the hand rope, the effect being to screw the sheave *B* slightly away from *C*. This allows *C* to revolve freely on the shaft, and consequently the load will descend as long as the motion of the hand rope is continued, but, as soon as this motion stops, the sheave *B* will become jammed tight against the collar *C* and all motion will cease.

# BELT-POWER ELEVATORS

**Types.** Elevators operated or driven by leather belts first utilized factory-line shafting as a source of power. They may be classified as either spur- or worm-geared. In making this statement it is done advisedly, for, while there are doubtless others, it is safe to say that these two types are the most common and that other types are usually isolated cases. Both types of machines are made for floor and ceiling mounting, and may be located in the basement or at any story of a building.

## SPUR-GEARED MACHINES

The spur-geared machine, as can be seen in Fig. 37, is simply a train of spur gears and pinions having pulleys for leather belts on

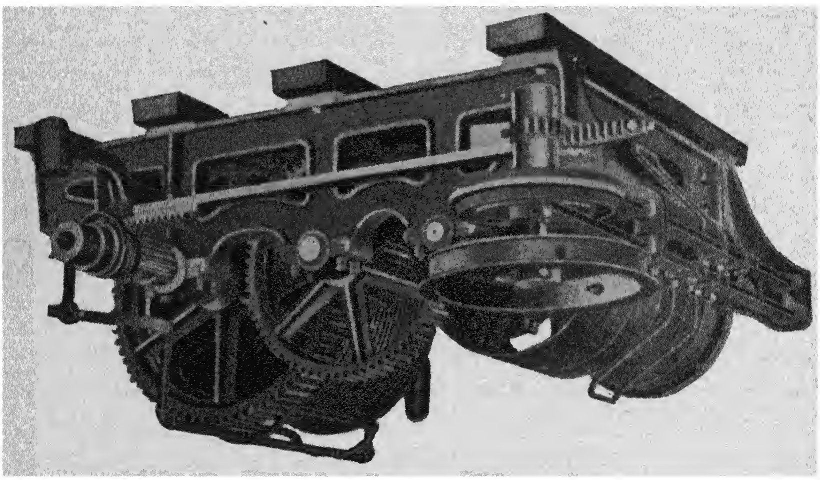


Fig. 37. Ceiling Type of Spur-Geared Belt-Power Elevator

the first shaft and a spirally grooved spool or drum for winding up the hoist rope on the last shaft. A middle or intermediate shaft is not absolutely essential but is used for convenience. By its use smaller spur gears may be used on the drumshaft, and the whole machine may then be made less cumbersome.

**Gearing.** These three shafts have journals at each end. These run in pillow blocks or in babbitted bearings usually cast in one piece with the side frames which serve as the support of the entire machine. The pinion, which is keyed on the first or pulley shaft, and the gear on the intermediate shaft are cut gears; that is, they have their teeth

formed in the gear cutter because the speed at which they run makes this neccessary in order to reduce the noise in operation, as well as to lessen the vibration which would ensue from the use of rough-cast gears. To further insure the safety and life of the machine, the pinion on the pulley shaft should be cut from a cylindrical piece of wrought mild steel which has been previously bored in the lathe to fit the pulley shaft.

The cut gear is keyed securely on the intermediate shaft, and alongside it is keyed a cast pinion having about 14 or 15 teeth, $1\frac{1}{4}$-inch pitch, and 4-inch face. This pinion in turn drives a gear having about 75 teeth, which is keyed on the drumshaft as well as bolted to the drum. The diameters of the shafts are as follows: first or pulley shaft $2\frac{5}{16}$ inches, intermediate shaft $2\frac{7}{16}$ or $2\frac{9}{16}$ inches, drumshaft $3\frac{3}{16}$ inches or $3\frac{7}{16}$ inches, as convenient to obtain. When the gear is bolted to the drum, it relieves the drum shaft of torsional or twisting strain.

**Pulleys.** The pulleys used for a machine of this capacity can be 20 to 22 inches in diameter and suitable for 4- to 5-inch belts, according to the load it is desired to lift, whether it be 2000 or 3000 pounds. Three pulleys are employed, two of them being idlers or loose pulleys, and the middle one a tight pulley, that is, keyed firmly to the pulley shaft. The loose pulleys are usually bushed with bronze sleeves and are always provided with compression grease cups to properly lubricate them. These pulleys are kept in place against the hub of the tight pulley by collars and set screws. Each shaft is provided also with collars to run against the ends of the pillow blocks to prevent end motion in the shafts.

**Belts.** The machine is driven by two leather belts from a "straight-faced" pulley on a line or countershaft. The width of the face of this pulley is equal to the combined width of the two loose and one tight pulleys on the machine. One of the leather belts is an open one and the other a cross belt for the purpose of obtaining reverse motions. The diameter of the pulley on the line depends on the speed of the line or countershaft, but it should be of a diameter capable of producing the speed desired at the machine. For example, suppose the winding drum is 30 inches in diameter, or 94 inches in circumference, and the desired speed of platform 50 feet per minute. About 6.4 turns of the drum per minute would be required to develop this

speed. Now the ratio of the gears and pinions is respectively 6 to 1 and 5 to 1, making the ratio between the first pinion and last gear 30. Hence the pulleys on the machine must revolve at the rate of 30 times 6.4, or 192 revolutions per minute.

*Belt Shipper.* The motion of the tight pulley in either direction is obtained by shifting, or "shipping", one or the other of the belts on to the tight pulley by means of the belt-shipping device, comprising two shipper bars fitted with belt forks or belt shippers and operated independently of each other by a cam movement, which will be discussed separately at the proper time. This cam movement is so arranged that when it ships both belts onto their respective loose pulleys, it simultaneously applies a powerful brake either to the tight pulley on the pulley shaft or to another special pulley on the same shaft. When the cam is moved to ship either belt onto the tight pulley to start the machine, it releases this brake, but not until the belt has a firm grip on the tight pulley, for here is the danger point in the spur-gear machine. This danger is greater when the lowering belt is shipped onto the tight pulley. Should the brake be released a moment too soon in starting, or too tardy in stopping, a sudden drop of a foot or two at the platform will result, especially if the load on the car is heavy.

The reason why the car or platform drops only a foot, or at the most two, before it recovers its self-control is due to the continued motion of the belt shipper after the operation of starting the elevator is begun. Should the movement of the belt shipper cease at this critical time, there will be nothing to prevent the car running rapidly down to the lowest point of its travel. Of course, the sudden arrest of the descent of the car by the belt being shipped over onto the tight pulley far enough to control it throws a great strain on the belt, which has been known to part, thus allowing the car to rush rapidly down to the lower end of the run.

**Centrifugal Safety Devices.** *Pulley Governor.* In order to provide a safety speed control the centrifugal governor, as shown in Fig. 38, was devised. The centrifugal governor includes a case or box of circular shape turned true on the inside and bolted to one of the side frames of the machine so as to be central and square with the pulley shaft, which is made long enough to reach almost through this case. A spider, or hub with arms, to which are pivoted levers

or weights, is also used. These weighted levers are provided with rubbing shoes, usually faced with either leather or hard maple, to rub on the inside of the rim of the box or shell case. The levers are held back from rubbing on the shell by springs, the tension of which is adjusted to the speed at which the pulley shaft is to run. Any increase of speed beyond the normal tends to throw the shoes against the sides of the case, producing friction which retards the motion of

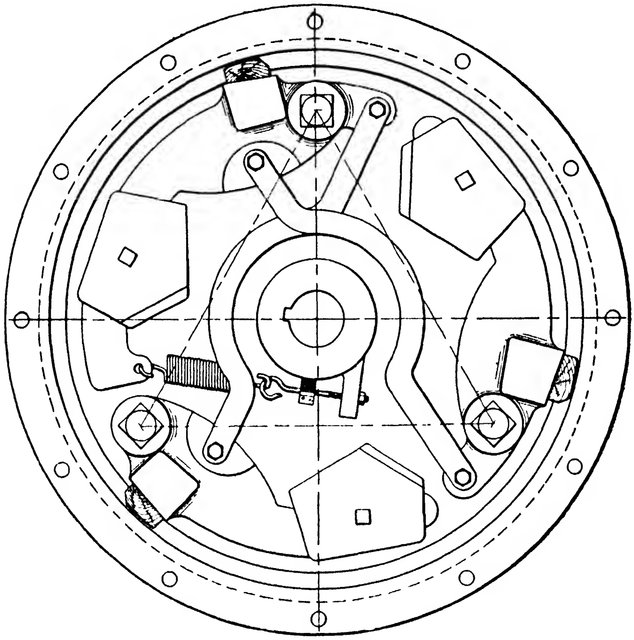


Fig. 38. Side View of Pulley-Type of Governor

the machine but does not stop it, thereby permitting the car to run down but not as quickly as without the governor.

*Drum Governor.* One of the oldest and a good form of governor is shown in Fig. 39. It comprised a winding drum placed overhead in the hatchway in place of the top or overhead sheaves generally used to lead the cables from the winding machine to the car. This drum was scored with spiral grooves to receive the cables. Two cables were used. One was fastened to the drum on the winding machine and led to the drum over the hatchway, where the other end was fastened. The other cable led from this overhead drum down

to the car, the ends of the cable being attached to the drum and the car. The length of the first cable had to be sufficient to reach from the drum on the machine up to the overhead drum and around the latter as many times as were equal to the travel of the car, plus a few spare turns on each drum. The length of the other cable was from the overhead drum down to the top of the car, when in position at the lowest landing, plus two or three turns around the drum.

The drum at the top of the hatchway had cast on the end of it a very broad band or pulley $A$, which was turned true and smooth

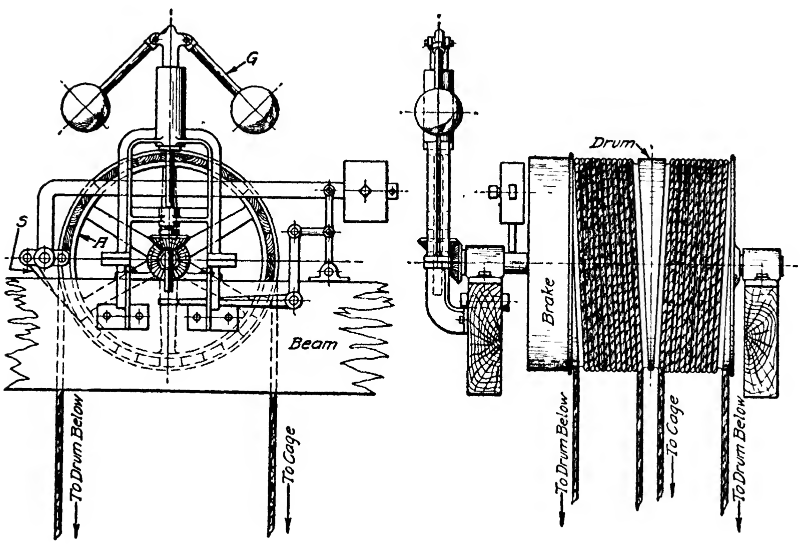



Fig. 39. Side and End View of Drum-Type of Governor

in the lathe. Around it was bent a band of steel $S$, lined on the inside with blocks of maple and fitted with a powerful wrought-iron lever, at the end of which was a heavy cast-iron weight. The whole apparatus formed a very powerful brake. Instead of a cam being used to release the brake or apply it, it was held inoperative by means of a bolt or catch which, when the lever was raised, kept it there, the brake being off when in that position.

On the end of the drum shaft was a miter gear, which drove a similar gear on the frame of a two-armed governor $G$, the vertical spindle of which, instead of opening and closing a valve, would, in

case the drum revolved at an abnormal speed, release the bolt and allow the lever and weight to drop, thus applying the brake.

When the hoisting belt was shipped on to the tight pulley, the drum or the winding machine commenced to wind up the cable connected to the upper drum which, being already wound on that one, would in the process of unreeling cause it to revolve. It in turn wound up the cable attached to the car, thus hoisting it up the hatchway, and *vice versa.* If, when the motion was changed for lowering, the cable between the upper and lower drum should break, the quick descent of the car would cause the upper drum to revolve fast enough to make the governor trip the bolt and let the lever drop and apply the brake.

This safety device, however, had the defect of being incomplete, inasmuch as it was operative only in case of a rundown owing to a broken or imperfectly shipped belt, or when the cable between the drums parted, but not when the cable from the drum to the car broke. This latter contingency was provided for by the use of safety dogs on the car, which were actuated by a spring.

This safety device has been described in detail because it was at one time considered one of the best in use, and because it was the forerunner of the modern type of governor safety. This form of safety was introduced somewhere around 1864 or 1866, when power elevators were in their infancy.

**Undesirable Features.** The difficulty of obtaining a suitable safety device with spur-gear elevators is the strongest objection to this type of machine. This, combined with the fact that the operation of such a machine is noisy, often overweighs the fact that this type of machine is at least 85 per cent efficient.

## WORM-GEARED MACHINES

**General Design.** Turning now to worm-gear apparatus, Figs. 40, 41, and 42 give a general idea of the appearance of the ceiling and floor types of this winding machine. The gearing is enclosed in a cast-iron box or casing, usually termed the shell. This casing is used to confine the oil used for lubrication, and also to serve as a receptacle or reservoir for it, as this form of gearing requires a constant and very liberal application of oil. The worm is almost universally placed below the gear, partly because the machines are

hung to the under side of the floor joists on cleats similar to the way shafting is hung, but mainly so as to allow the worm to run immersed in oil.

**Parts and Their Mountings.** The apparatus consists of the worm, which is simply a short screw of coarse pitch, with the shaft usually an integral part of it; a toothed gear in which the worm runs; and a shaft to which the gear is keyed firmly. This shaft is made long enough to extend through the bearings in the shell or casing, and also to allow the keying on one end of the drum or spool on which the hoisting rope or cable is wound and the placing of a journal beyond

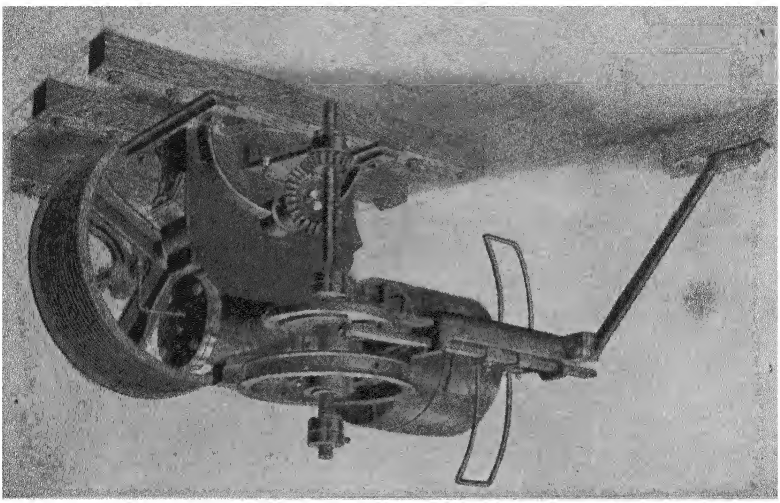


Fig 40. Typical Ceiling Type of Worm-Geared Belt Elevator with Plate-Cam Belt Shipper

the drum for support. This shaft is called the "drum shaft". A hanger with babbitted bearing, or box, for the outer end of the drum shaft to run in is provided, being either cast on or bolted to the shell or gear casing. The bearings for both the worm and drum shafts are provided with pads or patches to which are bolted the attachments for the limit stops, the belt shippers, two loose pulleys and one tight pulley, the usual belt-shipping apparatus, the automatic or limit stops, and a hanger for the support of the outer end of the worm shaft.

**Operation.** In order that the student may obtain a clear understanding of the operation of the worm and gear, a section of the shell

or housing is given in Fig. 43. When the worm, which is shown in cross section, is caused to revolve on its axis, the spiral thread pro-

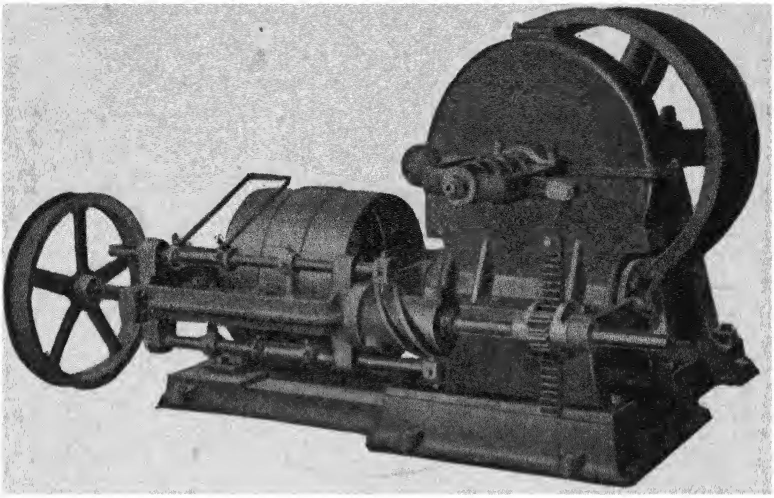


Fig. 41. Floor Type of Worm-Geared Machine with Cylindrical-Cam Belt Shipper

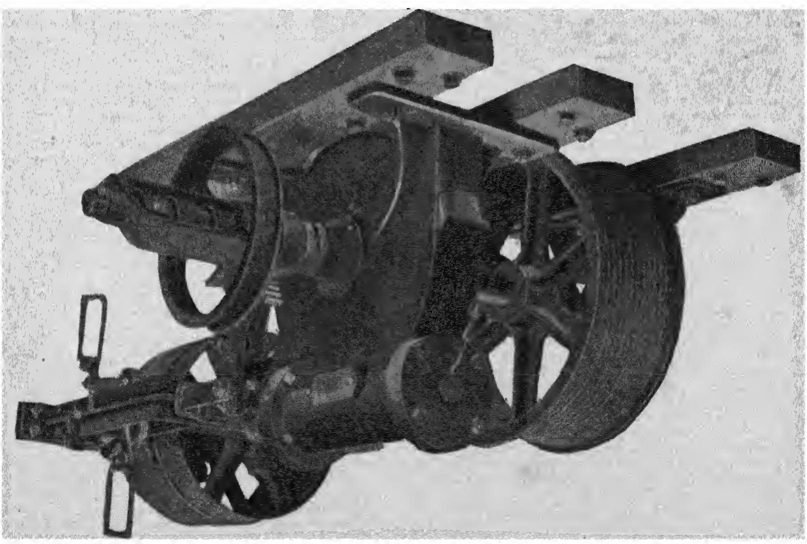


Fig. 42. Ceiling Type Worm-Geared Machine

duces an endwise motion similar to that of a screw in a nut; but, as the worm is prevented by its bearings from moving endwise, it transmits that motion to the teeth of the gear and causes it to revolve.

This motion is transmitted through the drum shaft to the drum, which winds and unwinds the hoist cables.

**End Thrust on Worm Shaft.** It is obvious that a heavy load on the hoisting rope will produce an intense pressure endwise on the threads of the worm, that is, endwise on the worm shaft. To illustrate, let us suppose that a load of 2000 pounds is being lifted, and that the drum is 30 inches in diameter and the worm gear 24 inches at the pitch line. Then the end thrust of the worm shaft will be $\frac{2000\times30}{24}$ or 2500 pounds. The dimensions given for gear and drum are about what are used in general practice, the drum being always

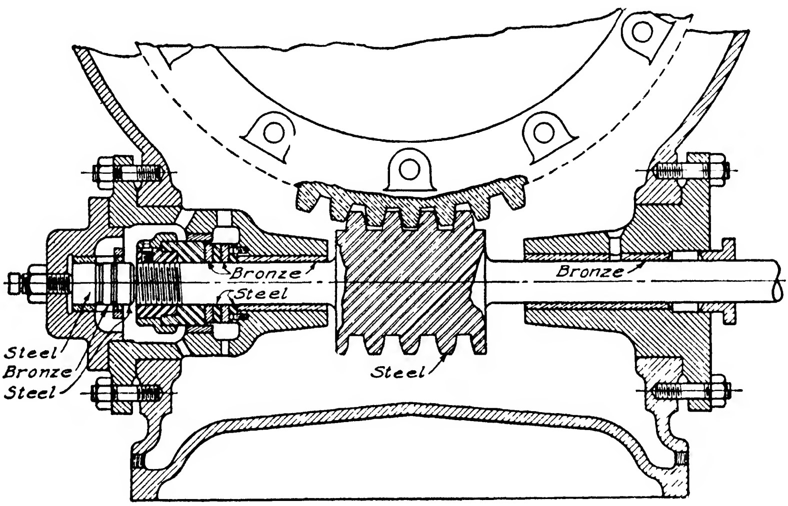



Fig. 43. Section through Worm-Gear Housing, Showing End-Thrust Arrangement

larger in diameter than the gear. This end thrust on the worm shaft is taken care of by means of what are called thrust buttons, which will be described later under the discussion of the worm and gear.

**Desirable Features.** *Locking Action.* The greatest advantage of using worm-geared machinery for elevator drive lies in the fact that, although the worm and gear unit permits motion to be transmitted from the worm to the gear, it does not permit motion to be transmitted from the gear to the worm. Hence worm-gear machinery is self-locking, for, since the winding drum is connected to the worm gear, it is impossible for any change in load on the car to cause

movement, because to do so the gear would have to be capable of producing motion of the worm. This feature is a very desirable one, because it means that no car-locking device is required, for the car cannot descend as the result of any load being placed upon it.

**Undesirable Features.** *Friction.* The spur gear, it will be remembered, generally consumes in friction less than 15 per cent of the energy supplied it; but a badly proportioned worm and gear may waste in friction as much as 50 per cent of the energy supplied, and will therefore be only 50 per cent efficient. But a well-proportioned worm and gear is not so uneconomical as this, for gears can be so built as to consume in friction 30 per cent or less of the energy supplied them.

*Worm-Shaft Speed and Pressure.* Reference to Fig. 43 will show that one revolution of the worm shaft will only cause the worm gear to move through the space of one tooth, while one revolution of the pinion of a spur-geared machine will cause the spur gear to move through as much space as is occupied by the same number of teeth as are on the pinion. Assuming, as was done in the case of the spur-geared machine, that the winding drum is 94 inches in circumference, and that a car movement of 50 feet per minute is desired, it will be found, if a worm gear having 50 teeth is employed, that the drum will make approximately 6.4 turns per minute, and that the worm shaft will have to revolve at the rate of $6.4 \times 50$, or 320 revolutions per minute, as required under the assumptions made in the case of the spur gear.

Mention has already been made of the large-end thrust on the worm-gear shaft. This also means that there is a large tooth pressure greatly in excess of that common with spur gears used for the same purpose.

## PRIME MOVERS

**Use of Separate Prime Movers.** Although these belted machines were originally built to meet the requirements of an elevator operated from a line of shafting in a factory, it later became apparent that there were many cases where, owing to the absence of power machinery, an engine would have to be employed for the sole purpose of operating the elevator. In such cases a steam engine, gas engine, or electric motor may be used.

**Steam Engine.** When a steam engine is employed it must be of ample power, that is, one rated at 150 to 200 per cent of the actual horsepower required. In all cases where the engine is to drive the elevator as its only, or as its principal load, the precaution must be taken to use a flywheel twice as heavy as would be required by the same engine when running constantly under a full or nearly full load. The reason for this is that the work of an engine driving an elevator exclusively fluctuates to such a degree that no governor has ever been made which can take care of the engine under such extremes.

For example, let us suppose that the engine is running with the elevator at rest. In this condition the engine is driving only the countershaft and pulleys to which the elevator apparatus is belted, the power delivering probably not more than one and one-half to two horsepower. In the meantime the elevator, having been fully loaded, is started; that is, the hoisting belt is shipped onto the tight pulley, an operation occupying not more than two seconds of time. Now, if the load requires eight horsepower to lift it, the engine is suddenly called upon for eight additional horsepower, with the result that the engine nearly stalls or stops until the governor opens up the valve and the engine takes steam and gradually recovers speed. However, in the meantime the elevator will have run nearly a story. When the car arrives at the stopping place and the belt is shipped back onto the tight pulley, the engine races for a few seconds until the governor has time to act. This kind of service is, of course, unsatisfactory, but it may be easily avoided by following the recommendation of having an engine of ample capacity equipped with a heavy flywheel.

**Gas Engine.** The trouble just mentioned is especially serious with gas or gasoline engines. This kind of engine rarely develops the full rated power and it cannot recover itself as quickly as a steam engine because it has to take in a charge of gas and air and compress it before an explosion can occur. Thus a gas engine is more liable to stop under these severe conditions than a steam engine. However, by installing an engine with a capacity of 150 to 200 per cent of the horsepower required and adding a flywheel of double the usual size no inconvenience will be experienced.

**Electric Motors.** *Non-Reversible Type.* In the case of driving the elevator with an electric motor no trouble of this kind will be experienced if the motor is shunt wound and greater than the capacity

required; but should the motor size be close to the actual horsepower required, lowering with a full load, especially if the elevator be of the spur-gear type, may cause the motor to race. In fact, unless the field of the motor is ample for its horsepower, the motor will be driven by the elevator at from 10 to 30 per cent above its normal speed, with the result that the motor will act as a generator. Hence, the obvious and most simple remedy in cases of this kind is to use a motor of ample horsepower and with properly proportioned fields both as regards volume of core iron and field copper.

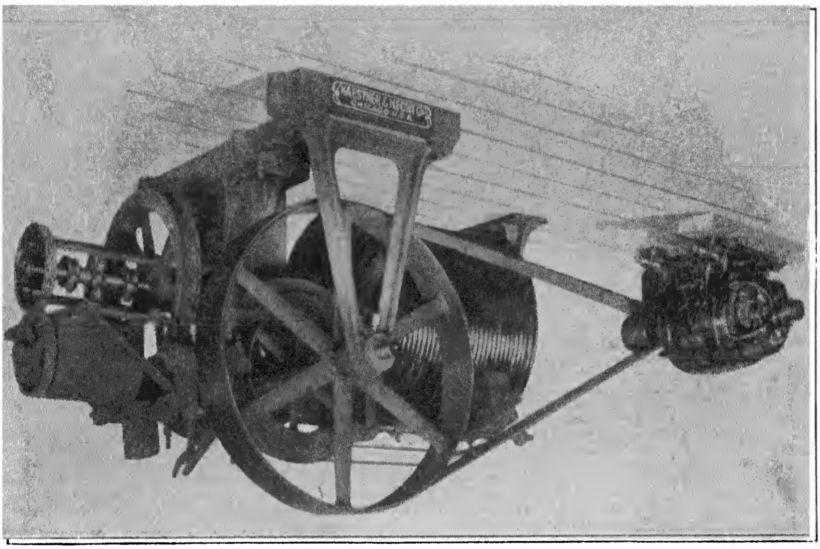


Fig. 44. Reversible Motor Drive for Belt Elevator
*Courtesy of Kaestner and Hecht, Chicago, Illinois*

In the case of a motor driving a countershaft a flywheel is of no benefit, as its momentum accelerates rather than retards the speed of the motor. Short-circuiting the armature through resistance coils will produce very satisfactory results, but the difficulty surrounding its application to this particular type of machine makes it impracticable.

*Reversible Type.* The belt elevator is sometimes driven by a reversible motor, which, although high in initial cost, possesses many advantages over the motor-driven countershaft and reversing-belt method. In such cases a compound-wound reversible motor is used

with a controller, Fig. 44. The hoisting apparatus is driven by a belt and pulley from the motor, which is hung to the joists in line with the hoisting apparatus. The motor is attached to a sliding base so that, as the belt stretches, the motor can by means of screws be moved to take up the slack in the belt. Such an elevator is called a single-belt machine to distinguish it from those using two belts and a belt shipper to obtain reverse motion.

In this type of machine the entire operation of the elevator is accomplished by stopping, starting, and reversing the motor by means of the controller, which is also provided with a mechanical connection for releasing and applying the brake. In some few cases the brake is applied by a powerful spring and released by an electrical connection which energizes a solenoid properly attached for the purpose, but this form of brake is not used extensively with the single-belt elevator. Braking action can also be obtained by short-circuiting the motor armature, as will be described in the discussion of modern electric-elevator control.

## SAFETY DEVICES

**Center Line.** Every freight elevator, especially those operated indiscriminately by anyone who needs the elevator for the time being, should be provided with a center line, which will be described later. In the hands of the unskillful operator it is essentially a safety device as well as a convenience. The operator has only to remember that to stop the elevator he must pull the center line until the car stops.

**Slack=Cable Stop.** The striker, which is part of the automatic stop, is a wrought-iron arm attached to the car having an eye encircling the operating cable and moving with the car during its entire travel in either direction. At each end of the run, a knob of cast iron called a stop button (see Fig. 81, Part II) is affixed to the operating cable, which should always be so arranged as to make a down pull on the rope correspond to an upward movement of the car, and *vice versa.* The button is made in halves, with a properly formed groove to fit over the cable, and is clamped tightly on the cable by means of bolts. The striker arm in traveling with the car will pull the operating cable back to its former position and thereby stop the elevator on meeting with an obstruction in the form of the

stop button. It is, in fact, an auxiliary limit stop and, if it is set properly, should stop the elevator of itself a little in advance of the automatic limit stop, which will be described later. It thus saves that valuable and useful device from wear and tear, leaving it in fine condition to perform its duties without fail, in case the operating cable breaks or the buttons slip. The student will readily see from this description that this combination really forms a double safety. The slack-cable stop is an attachment designed to stop the elevator instantly in case the cables become disarranged on the drum, and is an appliance which should be on every elevator driven by power.

## BELTS AND BELT SHIPPERS

**Belts.** Leather belts of double thickness and of full width should always be used. Cotton, rubber, or any other belt with a textile fabric for its basis is unfit for use as a shifting belt—that is, one which has to be shipped from one pulley to another. The action of the belt shipper frays the edges of this kind of belt, wears it out, destroys its tractive powers, and soon renders it unserviceable. Moreover, this kind of belt cannot be made endless, and the necessary belt lacings or other fasteners make lumpy joints causing noise and having a bad effect on the machine. Leather belts are stronger, wear better and, although they stretch more at first than the other kinds, can be made endless without impairment as often as they need to be shortened; but they should always be of double thickness and of strictly "back stock", short-lap, and riveted.

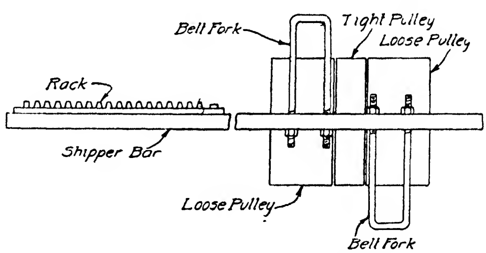


Fig. 45. Early Form of Belt Shipper

**Belt Shipper.** *Early Type.* In the first power elevators the belts were shipped or shifted from the loose to the tight pulley and the reverse, as shown in Fig. 45, by means of a bar of wood or iron sliding in suitable guides and moved lengthwise by means of a rack bolted to the bar, which was operated by a toothed pinion keyed on a rigidly supported short shaft. On this shaft was also keyed a sheave, around which the operating cable was wrapped two or three times and attached to the

sheave at the center portion of the wraps in order to insure its positive movement and at the same time to enable it to turn in either direction. Both belt forks or fingers, as they are termed, were attached to this bar, and it necessarily followed that both belts moved at the same time and together. The form of this device necessitated the use of loose pulleys of double the width of the belt, and, although the arrangement was simple, it was very clumsy in appearance.

*Cylindrical and Disk Types.* Appliances of this nature were devised later which shipped one belt at a time. A number of different devices were used, but the best and most popular were two in number, and it would be hard to decide which was the better, because each had desirable features not possessed by the other, and each was applicable to situations in which the other was not so convenient.

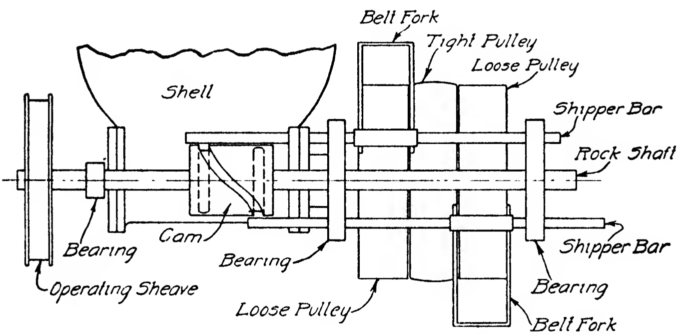



Fig. 46. Details of Cylindrical-Cam Belt Shipper Shown in Fig 41

Both were fitted with two independent shipper bars equipped with belt forks or fingers, between which the belt was made to travel. When a shipper bar was moved, its belt fingers carried the belt with it. As each shipper bar controlled only one belt, only one belt was moved at a time, the other remaining stationary while its companion was in motion. One of the awkward features of this device was that the shipped belt always had to be returned to its original position before the other could be shipped.

In both cases the operation of the shipper was accomplished by a cam. In one case the cam was a cylinder, Fig. 46, having a helix cast on its surface in the form of a groove into which a pin and roller worked. This is shown mounted in Fig. 41. Each end of this helical groove was

deflected from its spiral course and was made to run at right angles with the axis on which the cylinder revolved. The pins and rollers on the ends of the shipper bars were so arranged and set that when the belts were running on their respective loose pulleys, these pins were at either end of the helical groove. It is evident that if the cam were made to revolve in either direction, one pin would be caused to run in the helix while the pin on the other shipper bar would run in that part of the cam or groove which was at right angles with the axis. The pin running in the helix would be caused to traverse the length of the cam carrying with it the shipper bar, while the pin running in the groove at right angles to the axis would remain stationary.

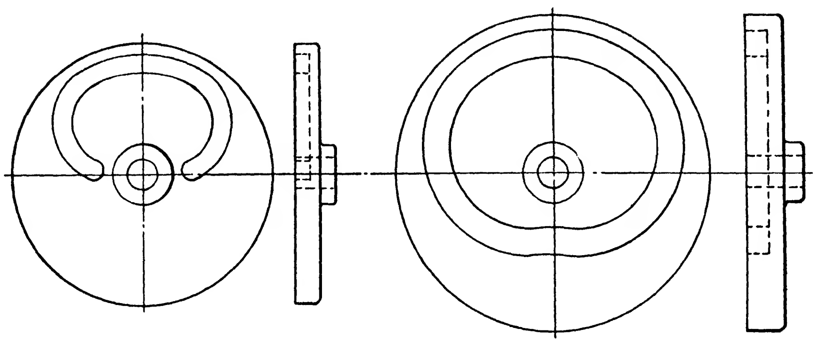


Fig 47. Details of Plate Cams for Belt Shipper, Shown in Figs 40 and 42

The other type of cam was a flat circular plate, having in it a groove or channel for the movement of the pins, Fig. 47. A portion of this groove was concentric with the axis on which the plate revolved, but at either end the curve approached the center of the plate, as shown. The device is also shown mounted in machines in Figs. 40, 42, and 48. Pins which were set in this groove in proper position would be moved in a manner similar to that described above.

*Braking Means.* A feature common to all three types was a cam for applying the brake. It was so shaped and arranged that, when the belts were both shifted to the loose pulleys, the brake cam pressed directly against the brake shoe or against the end of the brake lever to which the shoe was fixed. This caused the shoe,

which was of cast iron and lined with sole leather, to press hard upon the tight pulley.

Some makers use a separate pulley for braking, though no

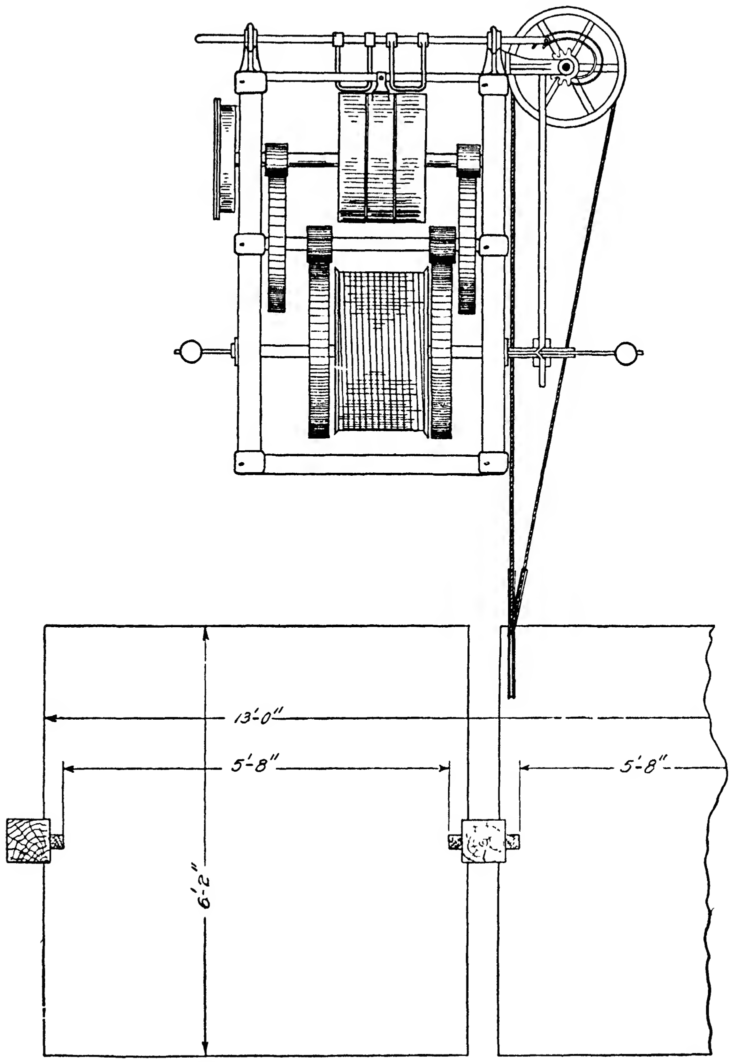



Fig. 48. Diagram Showing Application of Plate-Cam Belt Shipper

special advantage is gained by this except that with the independent pulley a brake band may be used, the latter arrangement being inapplicable to the tight pulley of the machine.

# WORM AND GEAR

**Historical.** Among the various forms of gearing used for the transmission of power, none is so little understood by those making and using them as the worm and gear. It may safely be said that, until the advent of the power elevator, about 1862 or 1863, the application of this form of gearing for use under great stress, and at what was then considered high speeds, had not been attempted on any large scale, and that its performance was not very satisfactory.

This was due to two or three reasons. One was the use of unsuitable materials in the construction, another was a lack of knowledge of the correct proportions of the members comprising this combination. Strange as it may seem, the very first attempts at its use were more successful than the efforts for improvement made later. This was due to the conditions attending both its production and use.

## PERIOD OF DEVELOPMENT

### EARLY TYPES

**Wrought-Iron Worms.** The first worms used for elevator service were made of wrought iron, owing to the fact that they were more easily produced in this way. In order that there might be no uncertainty about the strength and solidity of the threads, the quality of iron used was of the best. The worms were usually made by welding bands or rings of Swedish iron around a piece of shafting iron at the place where the worm threads were to be located. The whole was then put in the lathe and turned and the worm cut out of the solid forging at the place where the rings had been welded on. The use of Norwegian or Swedish iron was to prevent the possibility of there being faulty places in the threads of the worm—a condition which would impair its usefulness. The result was usually a clean, bright, close-grained worm of even texture and small diameter, all desirable qualities.

*Gear.* The gear was made of cast iron. In many cases it was made like an ordinary spur gear, but with the teeth set at an angle across its face to correspond with the angle of the threads of the worm. However, it was found this form of gear was not so lasting as another form made with a concave face. The latter form had the

advantage of presenting a greater surface to the worm and, as a result, was found to last longer and to be more efficient.

**Cast-Steel Worms.** Later, when steel cast ings came into use, many elevator makers began to experiment with worms made of steel castings, the threads being cast on the worm. The worm was then bored to fit the shaft and, after being keyed thereon, was turned, the threads being trued up in the lathe. In fact, before this time some experiments had been made with cast-iron worms, but it was found that a worm and gear both made of this metal did not work well together, although in some cases during the early stages of their use, where a smooth surface was obtained, the results were very satisfactory. In these isolated cases it was found that their satisfactory use was due largely to the kind of lubricant used, the lightness of the loads lifted, and possibly also to the closeness of the grain in the iron of which the worm was made. In general, however, the cast worm was not a success, but it did demonstrate the fact that the diameter and pitch of the worm had more to do with its successful operation than had been suppposed.

## CORRECT THREAD ANGLE

**Relation Between Thread Angle and Worm Diameter.** The first worms—those made by forging—were rarely more than $4\frac{1}{4}$ inches in outside diameter, or $3\frac{1}{4}$ inches at the pitch line, and of $1\frac{1}{2}$ inches pitch. Cast worms used on a shaft of the same diameter as the forged worms were made of 5 and even 6 inches outside diameter, or 4 to 5 inches at pitch line, owing to the necessity of leaving sufficient stock around the bore between the shaft and the root of the threads for the keyway. This, of course, changed the angle of the helix or thread as shown in Fig. 49. The outer parallelogram represents the outline of a worm 6 inches in diameter, and the inner parallelogram, of one 4 inches in diameter. The line $A$ is the center line or axis of the worm. The line

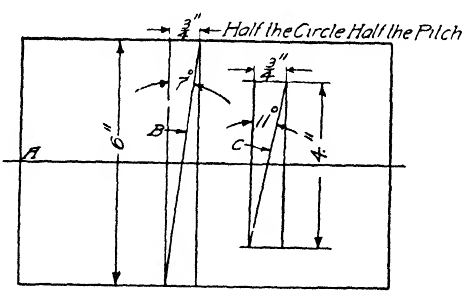


Fig 49 Variation of Angle of Thread with Worm Diameter, Pitch Constant

$B$ shows the angle of 7 degrees which the thread forms with a line perpendicular to the axis on a 6-inch worm, while line $C$ shows the angle of 11 degrees formed by the thread on a worm 4 inches in diameter and of the same pitch. It will be readily seen that the larger the diameter of worm, the smaller the angle between the thread and a line perpendicular to the axis $A$ of the worm becomes.

**Efforts Decrease Thread Angle.** It was thought at this time that a small angle for a worm thread was preferable, because in such a case the thread would be less liable to back down or reverse with a load. So firmly did this idea take hold that it was not considered good practice to use a worm having a thread which made an angle of more than 10 to 12 degrees with the perpendicular to the axis of the worm shaft. As the pitch of worm then used was always $1\frac{1}{2}$ inches, the only way to keep the angle small was to increase the diameter of the worm.

In their efforts to keep the angle small many makers used worms of 7 and even 8 inches diameter with the same pitch of thread and the same diameter of gear, making the face and teeth conform to the increased diameter and to the angle of the larger worm. These, in all cases, failed to give satisfactory results, as increased friction, heating, and cutting developed. While some beneficial results were obtained by varying the lubricant and decreasing the pulley diameter, the general opinion was that a worm of larger diameter than 6 inches was not desirable, although no satisfactory explanation of the fact was obtained.

**Development of the Correct Worm Thread.** Early in the seventies, William Sellers and Company of Philadelphia, Pennsylvania, manufacturers of machine tools and geared-belted elevators, applied the worm to machine tools with little success. They immediately began to experiment on the subject and made some important developments. These experiments were not made public at once, and in the meantime developments were taking place among the builders of elevators.

*Causes of the Developments.* The introduction of the gas engine and a frequent demand for higher speed with the belted worm-gear elevator were the principal factors in bringing these developments about. The gas engine very seldom developed its rated horsepower in the early days of its existence, and among the schemes devised to

help it was the application of a heavier counterbalance weight. This was attached to the drum in such a way as to pull counter to the car and load. Usually it was made 500 to 600 pounds heavier than the car, and while it certainly helped the engine to lift a heavier load, this extra amount of counterweight had to be raised in lowering the empty car. It was also found beneficial in lowering loads because it assisted in preventing racing or running down; but it was soon discovered that the worm, which set up undue friction or heating, could not be used with a gas engine whose capacity was not in excess of that required. At the same time a demand arose for a fairly speedy passenger elevator which would not be so expensive as either the hydraulic or the steam elevator of those days.

*Most Efficient Thread Angle.* This led to experiments along the line of increasing the efficiency of the worm and gear, and the investigations then made led to exactly the same results as those made with the planer drives, namely, what is now accepted as a well established fact, that a worm for elevator service gives the best results when it is made of such a diameter and pitch that the angle formed by the thread with the perpendicular to the axis of the worm is from 15 to 20 degrees. It was also learned that the percentage of loss by friction is much less with a worm driven at a high speed, provided the angle of the thread is within the range mentioned or, at least, not below 15 degrees. A greater angle than 20 degrees does not produce any bad results; but at the same time it does not appear to increase the efficiency of the worm to any noticeable extent, while for elevator service it possesses the undesirable feature of not being self-locking, that is, with a heavy load the gear is capable of turning the worm. On the other hand, the worm of from 6 to 10 degrees of angle gives very bad results. Although it must run comparatively slow, and the lubrication must be of the best, the percentage of power wasted is high. Moreover, if the oil used is lacking in body, the lowering of a heavy load with a worm and gear of this description is accompanied by a loud screeching noise, which is very unpleasant, and any increase in speed increases all these troubles.

*Double-Thread Worm.* Some of the conclusions arrived at were disclosed by the efforts made to run worm-gear machines at a high speed with the worm of low angle. It was then that, in addition to the heating, another trouble developed—that is, the difficulty in

stopping the elevator gently with a high belt speed. To overcome this a worm of greater pitch was used. The pitch was increased from $1\frac{1}{2}$ inches to 3 inches, using two threads on the worm instead of one. This scheme retained the original thickness and depth of thread, but increased the angle, and, while it doubled the speed of platform, it did not necessitate an increase of belt speed. It was also noticed that much less power was required to drive the double-thread worm than the single. The stops made at the landings, however, were never satisfactory, owing to the fact that the brake had to be applied harder and more quickly because of the tendency of the worm to reverse as the result of the pressure caused by the load; in other words, by the tendency of the gear to drive the worm.

**Value of Experiments.** However, the belt-power elevator, upon which these experiments were mainly performed, was destined to be replaced to a large degree by the more modern types. Much of the objection to belt-power elevators lies in the accuracy with which the brakes have to be applied in making a stop. In shifting a belt for this purpose from the tight to the loose pulley, the belt finally reaches a point where there is not enough of it in contact with the tight pulley to control it. It is at this point that the brake must be applied. If the load on the car is heavy, this moment arrives much sooner than it does when the car is being hoisted empty; and in the case of lowering, the critical moment arrives still sooner. However the brake may be timed and adjusted, it will always be applied exactly the same, both as regards the time and force applied to stop and hold. Only at slow speeds is it possible to adjust the belted machine so as to make a gentle stop. Hence the use of the belted elevator as a passenger elevator has not been a success, but the information gained by these experiments has been invaluable, especially in the application of the worm and gear to the electric elevator.

## MODERN WORM AND GEAR

During this period, in which the most advantageous angle for the thread was learned, numerous attempts were made to increase the serviceability of gearing. Henry Hindley of New York City introduced a worm which was longitudinally concave, and which was known as the Hindley gear, Fig. 50. Although applied to numer-

ous elevators it was found to be unsuccessful in that form, owing to the excessive friction or braking action which developed with wear.

**Introduction of Bronze Gear.** Bronze gears used with a forged steel worm were introduced during this period, thus giving us the most successful worm-gear combination of today. Only the rim containing the teeth is made of bronze, a flange usually being provided for bolting it to a cast-iron center forming the hub and web of the gear. Sometimes the flange is dispensed with and the ring containing the teeth is bored to be forced on the cast-iron center and afterwards pinned to keep it from moving. The use of a bronze worm with either

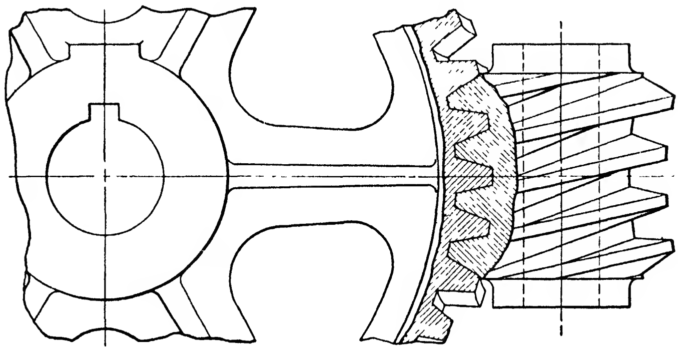


Fig. 50. Details of Hindley Worm Gear
*Courtesy of Albro-Clem Elevator Company, Philadelphia, Pennsylvania*

a cast-iron or a bronze gear was tried, but it was found that the worm wore out very quickly.

**Gear-Tooth Construction.** *Tooth Shape.* Various forms of teeth were tried out, but none of them seemed to produce any noticeable change. The tooth of the worm gear having a concave face cannot, of necessity, be straight across the face, the tooth tapering from the root to the point, and at the same time being thicker at the center of the face than at the outer ends. A tooth when viewed from the top is thus seen to have convex sides, the convexity being more pronounced at the point than at the root of the tooth.

*Method of Forming.* This shape is very difficult to produce in the gear cutter, in fact, only an approach to the proper shape can be made in this way. The best gears are first cut in a gear cutter and then finished with the hob. In cutting the teeth in the gear cutter three operations are necessary. First, the cutter is run in at the proper inclination with the face to produce a tooth at the angle

required by the pitch and diameter of the worm. The result is a parallel tooth. Then the gear is skewed around to the proper angle and the cutter is run in again so as to taper the tooth at one end. The gear is skewed back to the same angle the opposite way and the cutter again run in, producing a tooth tapered at the ends but having by no means a true concave face. To make this curve the hob is used.

*Details of Hob.* The hob is a worm cut in the lathe in exactly the same manner as the worm that is to be used in the gear to be hobbed. It is made of tool steel suitable for the purpose for which it is to be used; and, after being turned, cut, and polished, it is fluted similar to the way a tap for threading nuts is made, except that its sides are parallel. In fact, it is really a large tap, parallel in the body and with a stem at each end. The stem is made as short as practicable for the sake of rigidity. After being tempered, it is mounted between the lathe centers and driven from the face plate by a suitable driver or dog. The gear to be hobbed is mounted on a table bolted to the saddle of the lathe, and is made to revolve horizontally on a pin on the table, being fed by hand up to the hob by means of the crossfeed of the lathe saddle.

Hobs are sometimes made of cast iron with steel blades inserted to do the cutting, but, although cheaper to produce, they are neither so effective nor so lasting. Some shops have a special machine to do the hobbing, in which the table to which the gear is attached revolves with the gear and is driven by gearing connected with the hob, so that both run in unison. With this arrangement, gears can be cut from the solid material with the hob, and consequently a much better and more perfect gear can be made.

**Operating Characteristics.** *Pressure Distribution.* To understand the necessity for this shape of tooth and the distribution of pressure, refer to Fig. 43. The worm, it must be remembered, is cylindrical in shape with a helical tooth wrapped around it. In its relation to the gear the worm is tangent to its periphery. Now, during a revolution of the gear, the first points of contact with the worm are made at the extreme outer corners of a gear tooth, and it will be readily seen that the tooth must be thinner at this place because otherwise it would not enter the worm. For this very reason, and also because of the concavity of the face of the gear

the worm at this position of the tooth bears only at the two outer corners of the tooth. As the gear revolves, however, the points of contact gradually change toward the center of the face of the tooth until the gear tooth is fully in mesh with the worm, as shown by the tooth on the center line in the figure. At such a time the pressure is only at the center of the face of the tooth and, as the gear tooth passes this place and moves on until it leaves the worm, the point of contact is divided and gradually spreads until it is at the same two outer corners again when the tooth leaves the worm. It will thus be seen that the worm does not bear over the entire surface of the gear tooth at any time, but that the surface in actual contact is always quite small in area. Of course, the fact that several teeth of the gear are partly engaged at one time helps to distribute the pressure, but this distribution depends upon the accuracy with which the gear is made.

*Motion at Contact Points.* In a spur or bevel gear the motion of the teeth on the surfaces in contact closely resembles that of two rollers running together, but in the worm and gear it is very different. It is a compound motion made up of a sliding motion produced by the revolutions of the worm across the face of the gear, and a circular motion brought about by the spiral advance of the thread of the worm. The threads of the worm advance in a straight line parallel with the center of the worm shaft, so that the advance of the worm threads may be compared to the movement of a rack driving a spur gear. This complexity of motion between the small areas of contact, and their constant changing of position, not only make the pressure per square inch very great, but are conducive to friction and cutting.

**Lubrication.** On account of the friction the lubricant used should possess the power of resistance against being forced out from between the surfaces. Animal oils possess this quality in a more marked degree than either vegetable or mineral oils, the latter being the least valuable for the purpose. A good quality of lard or neat's-foot oil is the best that can be used if the service required is severe, and a small quantity of the best quality of lubricating plumbago will increase the staying qualities of the oil. The plumbago will coat the surfaces of the teeth and threads and, as no amount of pressure will remove it, it really lubricates. Beef or mutton tallow melted and stirred into the oil before introducing it into the housing containing

the worm and gear gives the lubricant greater body or power of resisting pressure. The proportions found to be best for tallow are from two to three pounds of tallow to a gallon of oil, and for plumbago, about a handful of plumbago for the entire quantity of oil used at one time. This quantity of oil varies with the size of the housing, for enough oil to nearly immerse the worm should always be in the housing.

Another good substance to use in oil for the purpose of giving it a body is white lead. It has the quality of resisting heavy pressure and will not cause trouble by drying after being mixed with the lard, or with sperm or neat's-foot oil.

Castor oil is, on account of its viscous qualities, very good, provided the service required is not too severe, but it has the objectionable feature of disintegrating when heated. When the service required of the worm and gear is such that it heats under heavy and continuous work, this oil first becomes thin and then separates into a watery fluid having no lubricating qualities and a thick, ropy, glutinous semi-solid, like India rubber in appearance, which sticks to the sides of the housing. An oil which fails like this at the very time a good lubricator is most needed is not to be recommended.

To resist heating nothing is better than flour of sulphur. This mineral is suggested because, having been sublimated, it is likely to be free from earthy impurities, and because it is in a state of fine division, a condition which is favorable to its working into the places where most needed.

**End-Thrust Blocks.** *Button Type.* About the year 1873, having met with considerable trouble on account of the heating and sticking of the thrust bearings at the ends of the worm shafts, the writer devised the form of end-thrust buttons referred to on page 46; this form proved very effective and serviceable, and is a favorite with most elevator builders today. This end-thrust device is very simple in construction, comprising merely several disks of metal, being alternately of tempered steel and hard, tough bronze, as shown in Fig. 51. They are made slightly smaller in diameter than the worm shaft, partly to allow free circulation of oil around them, but principally to produce an eccentric motion in order to keep the surfaces in contact constantly changing, and thereby lessening the liability to stick to one another. To further facilitate the flow of

oil between the bearing surfaces, an oil groove should be cut clear across the face of each disk, being careful not to have the grooves on the opposite sides parallel to each other, but rather at right angles. This precaution is taken to avoid weakening the disks, which should be from $\frac{3}{8}$ to $\frac{1}{2}$ inch thick when new, and which must be renewed after they have worn down to $\frac{1}{4}$ inch in thickness. Under ordinary conditions, this does not take place for a year or two—in fact, they have often been known to last for a period of four years before requiring renewal. The bronze disks always wear out more quickly than the steel. Should the thrust buttons, as these are called, require attention either from heating or cutting, or from

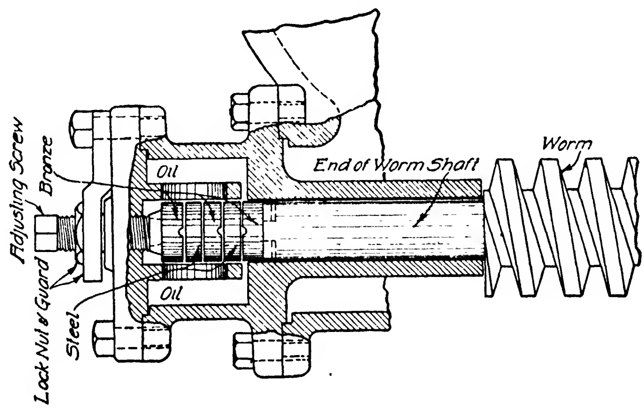



Fig. 51 Diagram of Worm Thrust Block, Showing End-Thrust Buttons

wear, they are readily accessible by taking out the back head. But it must never be forgotten that several precautions must be taken before doing this. The cage should be blocked, and the hoist cables should be made slack enough to remove the strain on them by backing the machine a little after the cage is blocked, while the oil should be drawn from the reservoir.

*Collar Type.* When the counterpoise weight in use is heavier than the cage, provision has to be made for thrust in the opposite direction when the elevator is running with no load. Then the counterpoise weight pulls the worm toward the front head of the shell. To meet this condition, rings of brass and steel are used, being slipped over the worm shaft and placed between the end of the worm and the end of the front head of the shell as shown in Fig. 52. The work required here being neither so constant nor so severe as with the

thrust buttons, one bronze ring is frequently found to be sufficient. It can be from $\frac{1}{2}$ to $\frac{5}{8}$ inch in thickness, and should be bored fully $\frac{1}{32}$ inch larger than the worm shaft in order to produce the eccentric motion while revolving, which tends to prevent the wearing of grooves in the wearing surfaces. For the same reason the outside diameter of the rings should not be larger than that of the end of the worm and front head, otherwise a shoulder will be worn in the ring which will in time defeat this object. The end of the front head must be protected from wear by facing it with a steel ring pinned on the end of the head, cast iron being a poor material for withstanding this kind of wear under presssure. In all cases where the amount of over-counterpoise is great, it is best to use three loose rings, the middle one being of steel, about $\frac{3}{8}$ or $\frac{5}{16}$ of an inch thick, and the two bronze ones $\frac{1}{2}$ inch in thickness. The steel rings at this end need not be hardened, the

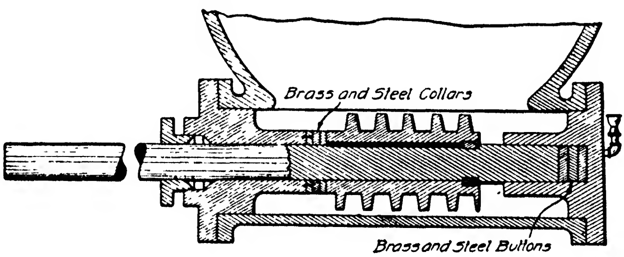



Fig. 52. Worm Thrust Block, Showing Use of Buttons and Collars

work to be done at this point being intermittent and never so severe as with the thrust buttons at the end of the worm shaft. The changing of the thrust from one end of the worm to the other, brought about by variations from no load to full load, introduces another condition which has to be taken care of, and which does not exist where the counterpoise is lighter than the cage. This is lost motion caused by wear. In the case of a counterpoise lighter than the cage, the pull is always in one direction, and any wear of the thrust is taken up automatically; but, in the case of the heavier weight, whenever a load is removed from the cage, the excess amount of counterpoise weight pulls the worm up against the rings and *vice versa*. When the load on the cage is in excess of the extra amount of counterpoise weight, it pulls the other way.

This matter of counterpoise may seem insignificant to the reader,

but in reality it is very surprising in its effects, for, the drum being always larger in diameter than the gear, the lost motion is somewhat magnified at the floor of the car and gives rise to the impression that something is loose and liable to give way. Moreover, the change of position of the worm is accompanied by a loud noise resembling a blow with a heavy hammer, the sound being produced inside the shell or gear casing. This is more noticeable when a load which about balances the amount of extra counterpoise is on the cage. In this case, when the brake is applied to stop the elevator, the noise is very pronounced. To remedy this trouble a set screw and lock nut are used in the back

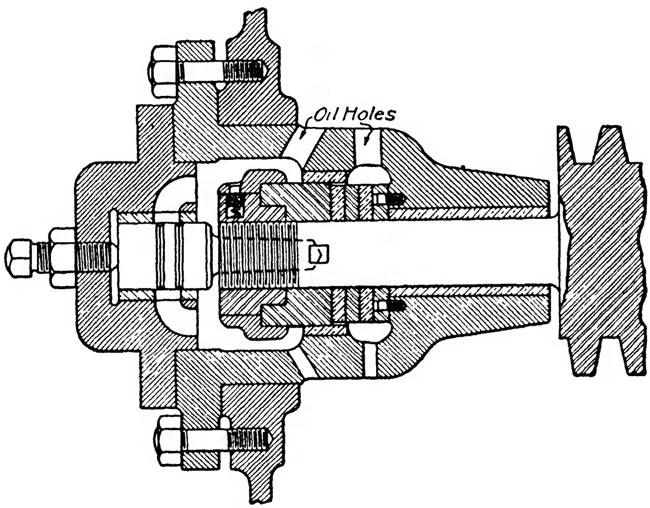



Fig 53. Taper-Plug Type of End-Thrust Block

head to move the back block forward, when required to take up wear. In adjusting this block, care must be taken not to set it up too tight or cutting will result. A small amount of play, about $\frac{1}{64}$ of an inch, is desirable and absolutely necessary to allow the lubricant to flow between the surfaces; but when the wear amounts to $\frac{1}{16}$ of an inch the noise becomes very loud. Under proper conditions, however, it requires many months to wear to this extent.

*Steel-Plug Type.* Several other forms of thrust device have been devised, some of which will be described here. One method was to bore the end of the worm shaft in a tapering manner similar to the nose end of a lathe spindle and fit it with a tapered steel plug, the end

of which was enlarged to the diameter of the worm shaft, Fig. 53. This steel toe, as it is called, was tempered and made to run against a tempered thrust block somewhat similar to the toe and step used in the spindles of the old burr-stone mills used for grinding wheat, and which doubtless inspired the idea for this form of thrust.

*Loose-Ring Type.* Another method was to use loose rings at both ends of the worm instead of at one end of the shaft. These, however, did not give the amount of wear expected of them and caused some trouble on account of heating. However, they are still in favor with many makers.

*Ball-Bearing Type.* The thrust block which seemed to give the greatest promise of good results was the ball-bearing, and, although it did prove very satisfactory at first, it was found that under light loads it was not durable, and that under heavy duty it failed entirely. The balls were of steel, one-half inch in diameter, and were usually inserted in a perforated plate about $\frac{5}{16}$ or $\frac{3}{8}$ of an inch thick. Holes were drilled in the plate, so that the balls did not run in the same circle, and a steel plate was placed on either side to take the pressure, the whole being located in a separate chamber kept supplied with oil. Under heavy pressure the balls would crush, and under fairly favorable conditions they would cut grooves in the plates or would wear flat places on themselves. As a result of these defects this form of thrust is very little used today.

**Worm and Gear Proportions.** The best proportions for a worm and gear for elevator service are as follows, based on the actual experience of some of the more prominent makers of elevators:

The diameter of worm should not exceed one-fifth the diameter of the gear, and the face of the gear should be about two-fifths the diameter of worm. These proportions are not arbitrary, but are approximately what have been found to give the best results.

The pitch of gears and worms for loads from 1500 to 2500 pounds should be about $1\frac{1}{4}$ inch; from 3000 to 5000 pounds, $1\frac{1}{2}$ inch, and from 5000 to 7000 pounds, $1\frac{3}{4}$ inch. The number of teeth in the gear does not differ widely, 45 to 50 teeth in the gear being sufficient for light capacities, and 50 to 54 teeth being suitable for high capacities. In fact, 50 teeth for any of them will give good results. The proper widths of belts and speed of pulleys will be considered under the head of horsepower.

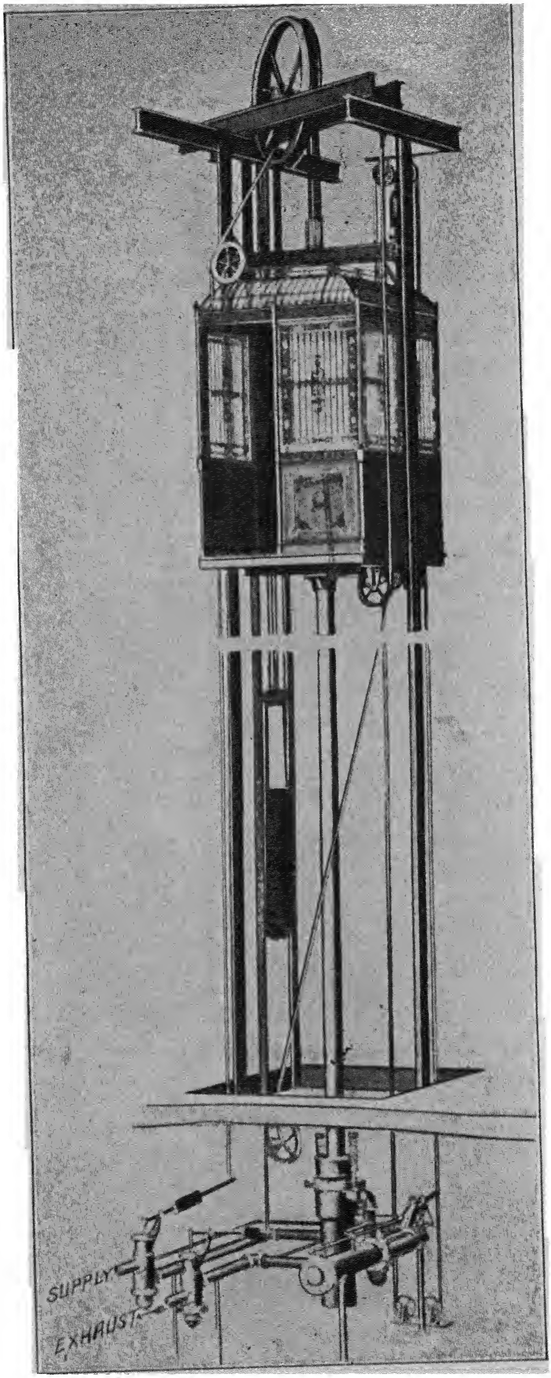
**The Tandem Worm and Gear.** Before leaving this subject, it is well to mention another form, or rather, combination, of worm and gear which has been in use for very many years and is today advocated by some makers of elevators, namely, the tandem worm and gear.

It comprises two worms on one shaft, situated far enough apart to allow the respective gears to clear each other. (See Elevators, Part III, Fig. 153.) The worms as will be seen in the diagram are of the sames pitch, but one is a right- and the other a left-hand thread, the gears being built accordingly. At the side of each worm gear a spur gear of suitable diameter to mesh with the other is placed. In ordinary use only one drum is used. This is connected to one of the gears, the other running as an idler, its office being to distribute the pressure.

The advantages derived from this form of gearing are, first, the elimination of the thrust bearing, the thrust being taken up between the two worms and equally divided between them; and, second, the reduction of the actual pressure between the tooth of the gear and the thread of the worm. In cases where the work to be done or the loads to be lifted are excessive, this arrangement has its advantages. In some cases it has been used as a duplex machine; that is, two sets of tandem gearing have been run side by side, each worm shaft being driven separately by its own motor, but with the motors coupled or connected to the same controller so as to be actuated in unison. To further insure uniformity of action, the worm gears were connected by spur gears meshing together. If this precaution were not taken, a slight movement of one worm shaft in advance of the other would have the effect of locking the machine; but arranged as described, the gearing works harmoniously, and the load or pressure on the teeth of gear and the threads of worms is divided by four.

The disadvantages are cumbersomeness and expense, for every part in it so far as relates to the worm and gear is duplicated. Additional parts are required, such as the spur gears, etc.

When first introduced, more than thirty years ago, spur gears were not used, but the worm gears were made as spur gears, with their teeth set diagonally to suit the angle of the worms, and these gears meshed into each other. This form of gear did not give satisfaction in point of durability and was abandoned for the present arrangement.




**DIAGRAMMATIC VIEW OF PLUNGER HYDRAULIC ELEVATOR**

*Courtesy of Standard Plunger Elevator Company, Worcester, Massachusetts*

# ELEVATORS

## PART II

## STEAM ELEVATORS

### HISTORICAL DEVELOPMENT

**Economy in Power No Desideratum.** The first direct application of steam to the operation of elevators, which occurred in the sixties, seems to have been, as far as economy of power was concerned, as successful as any subsequent effort. Later, more compact machines were devised, but all of them with one exception were very inefficient. In those days economy in power in the case of the elevator was not a desideratum; lifting power and speed were the objective points, and they were certainly obtained.

Up to the year 1858, the only elevators propelled by power were the worm-gear or spur-gear type. They were driven at a very slow speed by two belts from a line or countershaft and were used only for carrying freight. In that year the first passenger elevator was installed in the Astor House in New York City. It was one of the two-belted worm-gear type, and its speed was only 50 feet per minute.

**Retention of Gear Transmission.** It was only natural that, in applying steam direct, the worm and gear and the spur gearing should have been retained as part of the apparatus, just as it was nearly forty years later in the first effort to apply electricity to the same purpose.

*Worm-Gear Type.* At first a reversible, link-type, vertical two-cylinder engine with a 3-foot pulley on the crankshaft, as shown in Fig. 54, was used. The worm-gear winding apparatus was mounted on the floor of the engine room and provided with a 16-inch pulley for belting it to the engine, which was mounted on a separate foundation a few feet away. There was no need of a belt-shipping device, for the link-motion reversing gear on the engine performed the duties of controlling the direction of the running of

the engine, and hence of the elevator. In order to obtain the necessary stroke of the operating cable for controlling the car at a high

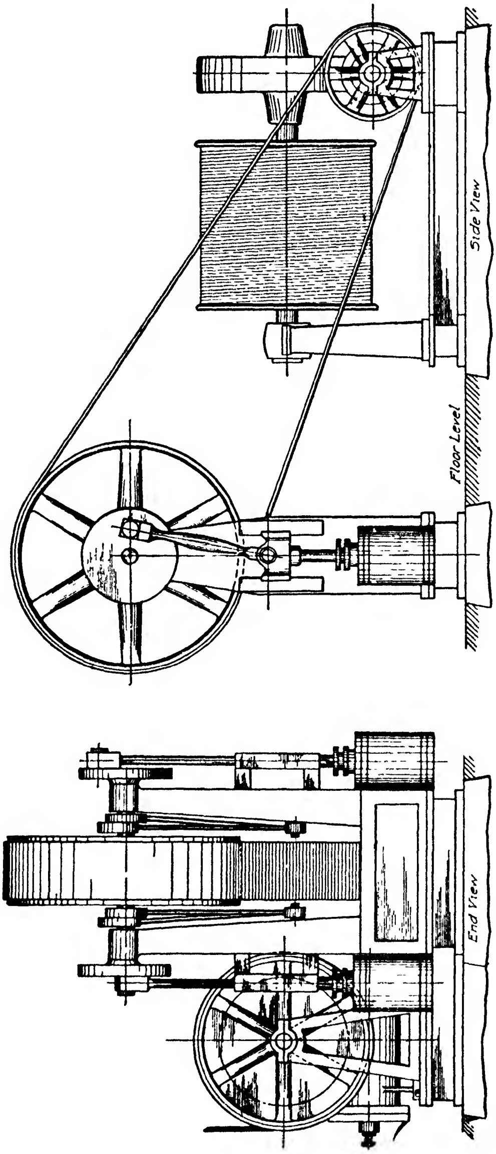



Fig. 54. First Steam Elevator. Belt-Driven from Engine; Worm-Driven Winding Drum

speed, a cut gear was attached to the rockshaft of the engine, this gear being operated by a pinion of suitable diameter. On the pinion

shaft was keyed a sheave around which the operating cable was wound two or three times and fastened. This arrangement gave a stroke of from two to two and one-half feet to the cable in either direction—a decided necessity for slowing down and stopping when running at a high speed. No brake was used, as all the worms in use with elevators at that time, and for years after, were of the single-thread type, in which the thread angle did not exceed 10 to 12 degrees. Hence it was not likely that the load would ever start

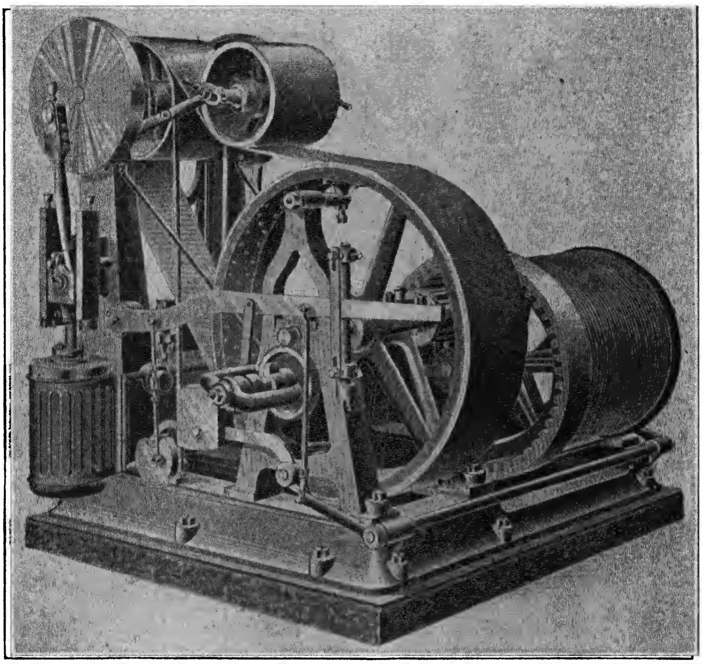


Fig. 55 Steam Elevator with Internal Spur-Gear-Driven Winding Drum

down from a state of rest, especially as it would have to drive the engine to do so.

This machine, while occupying considerable floor space, did not consume steam while the elevator was at rest and was very efficient, being a decided improvement over the old arrangement. But it was not long before efforts were made to design a more compact machine, and, while quite a number of contrivances resulted from these efforts, it is thought best to discuss only those which were successful and which stood the test of practical use.

*Spur-Gear Machine.* The spur gear was tried and, after many experiments, an engine with the steam apparatus, winding drum, and gearing all on the same bed, Fig. 55, was introduced and became popular. On the winding drum an internal spur gear and pinion was used, and the former order of pulley arrangement was reversed, the larger pulley being placed on the pinion shaft and the smaller one on the crankshaft of the engine. While the diameter of the latter was retained, that of the former was increased to four feet and necessarily fitted with a brake pulley attached to the arms inside the outer rim for, it will be remembered, a spur-gear machine is not self-locking. The pulleys were reversed and changed in size in order to reduce the driving ratio between the engine and the drum. This was necessary because the substitution of a spur gear for a worm gear increased the mechanical advantage between the pinion and the drum, owing to the fact that one revolution of a pinion causes a greater movement of a gear than one revolution of a worm used with a similar gear.

Another feature was necessarily introduced because, with the use of the spur gearing, it became unsafe to use the link motion as a reversing gear. Should the operator, when handling a heavy load, fail to throw the link motion completely, in the right position, the engine would not receive enough steam to handle the load and the car would back down against the steam; similarly, in lowering a heavy load, the car would run down too fast.

**Automatic Steam Valve Developed.** It was therefore necessary to devise a valve that would give the required amount of steam automatically and independently of the operator. Several types of valves were devised, the best and most successful of which will be here described.

*Otis Tufts Valve.* The Otis Tufts Company of Boston probably designed the first valve of this kind, Fig. 56, and it proved one of the best at that time. A sliding plate, *B*, was used between the distributing valves and the cylinder faces. This plate had two sets of ports with a blank space between, and it was arranged to slide at will at right angles with the axis of the cylinder. When placed in the central position, it completely closed the cylinder, preventing either ingress or egress, and in this way would stop the machine. When moved to one extreme of its travel, the ports were so arranged

that the distributing valve would let steam in directly to either end of the cylinder exactly as in an ordinary steam engine, the middle port being the port of egress or exhaust. The engine would run, of course, in the direction for which the eccentric was set.

When, however, this plate was moved to the opposite end of its travel, it reversed this order, for the steam which was admitted by

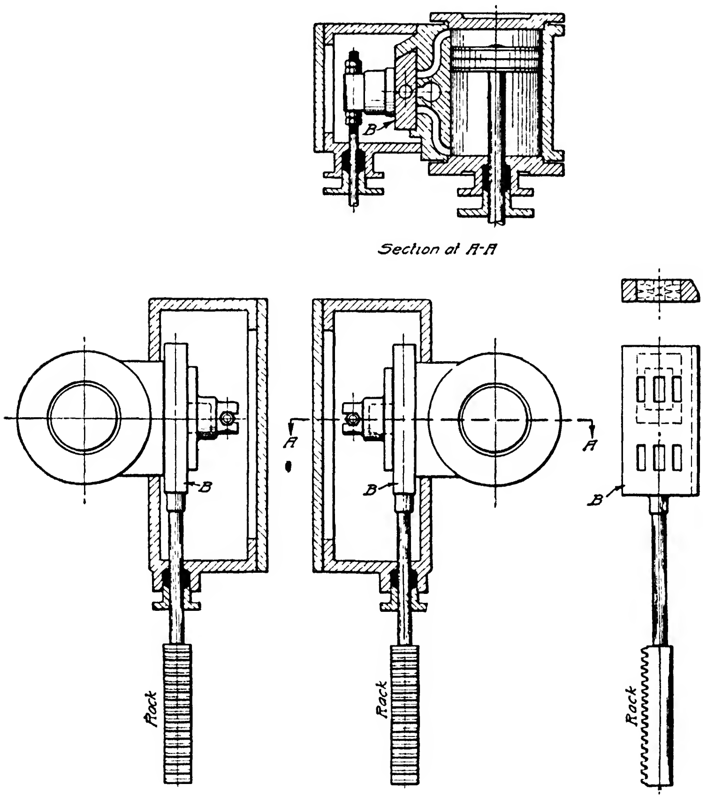



Fig. 56. Otis Tufts Automatic Steam Valve

the distributing valve to the upper port was carried down through the plate to the lower port of the cylinder and, after being admitted to the lower port in the plate, was conveyed to the upper port of the cylinder. By this means, a reverse motion was obtained without the use of the link motion and at the same time only one eccentric and rod were required. However, no lap or lead could be used on

the distributing valve, the stroke being always the same. This arrangement was not economical of steam but it was effective and safe. As in the case of the link-motion engine, a double-cylinder machine, with the cranks set at right angles and a steam chest common to both cylinders, was used and no throttle valve, the sliding plates just described performing the office of throttle valve, when placed in the central position. Steam was always on in the steam chest and, to prevent the access of water of condensation to the cylinders, the bottom of the steam chest was made two or three inches lower than the bottom port of the cylinders and a pipe led from the lowest part of the steam chest to a steam trap. In this way the trap removed water from the steam chest as soon as formed.

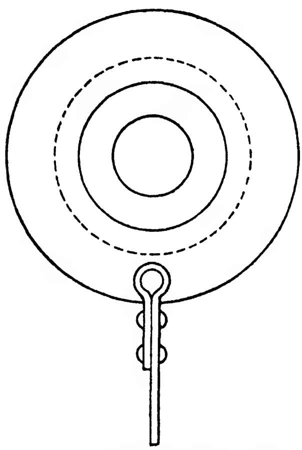


Fig. 57. Sheave for Steam Valve Mechanism

The movement of the plates in unison was effected by running valve stems from them out through stuffing boxes in the front of the steam chest. Each valve stem was fitted with a rack which meshed into a pinion on a rock-shaft. On the end of this shaft was fastened the sheave to which the operating cable was fastened and around which it passed. Hence a movement of the operating cable effected a movement of the plates and a corresponding performance of the engine. On this rockshaft was also keyed a sheave, a view of which is shown in Fig. 57. It was about 3 inches in diameter and had flanges on each side through which a pin was passed from side to side. It was keyed to the rockshaft in such a way that when the rockshaft was in the stop position the pin was at the top. To this pin was attached a leather strap which led up to and over two sheaves on the ceiling of the room where the engine was located. One of these sheaves was directly over the rockshaft and the other above the end of the brake lever, the strap being passed over these sheaves to the lever. If the rockshaft was revolved in either direction, it would wind up the strap and lift the lever, thus releasing the brake simultaneously with the turning on of steam to the engine. When the rockshaft was returned to the stop position and the strap

unwound, the brake was applied by a heavy iron weight on the brake lever.

*Miller Valve.* The Miller valve motion, shown in Fig. 58, was another device designed on lines similar to that of the Otis Tufts. It had the plate between the distributing valve and the cylinder face, but it was made circular in shape. In effecting the change in direction of the engine movement the plate was caused to revolve

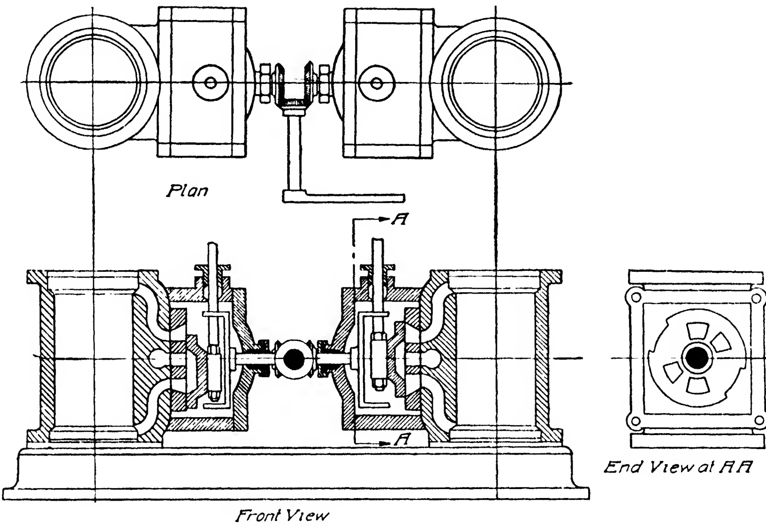



Fig. 58. Front Section, Plan, and End View of Miller Steam Valve

on a hollow gudgeon or pin cast on the cylinder face and acting as the exhaust port.

The brake in the Miller valve mechanism was released and applied exactly as in the case of the Otis Tufts movement. This valve gearing had many disadvantages, the principal one of which was the very contracted steam ports which its construction necessitated. Another was the necessity for two separate steam chests and an independent throttle valve which had to be operated from the rockshaft.

*Otis Valve.* The Otis Company of New York used two piston valves, as shown in Fig. 59. The piping or steam passages were cast in the back of the steam chest and met in the center of same, where a change valve was used to shut off the steam from the cylinders or change the direction of running as desired. Instead of causing the

distributing valve to act as a throttle, when in its central position, as in the case of the Otis Tufts valve, or of using a separate throttle, as with the Miller valve, a third piston valve was introduced. This performed the double office of changing the direction of the steam as it entered the engine, and also that of shutting off the steam when desired. These valves were fitted with rings similar to a piston and were closely fitted. The ports through which they admitted steam were cast slightly diagonal to prevent the rings from catching in their edges. This feature also had the effect of causing the valve to cut the steam off very gradually instead of abruptly. Their construction was such that they were always balanced; in other words, they were subject to no pressure from the steam—a feature

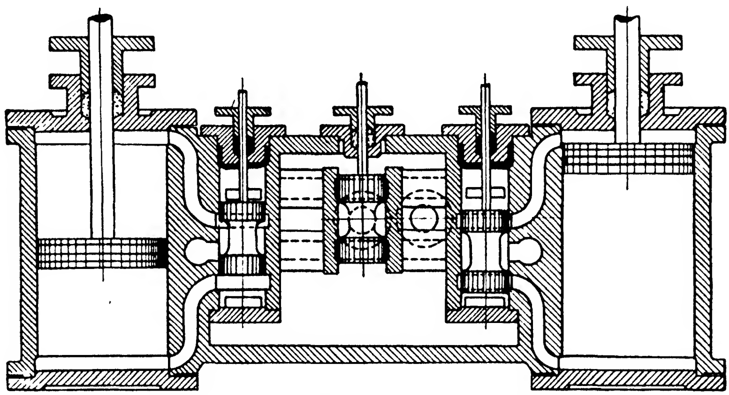


Fig. 59. Section of Otis Steam Valve

which both types of valve previously described did not possess. This elimination of the steam pressure from the back of the valve reduced friction, and, everything considered, the Otis valve was a decided improvement on the other two. Its chief defect was the difficulty experienced in keeping it tight, that is, in preventing the steam from leaking through it.

*Crane Valve.* In the latter part of the sixties another valve was introduced by the Crane Company of Chicago. This was really a simpler and a better valve than any of the preceding and it became quite popular throughout the West, although the Otis Company continued to make and use their valve until the steam elevator went out of use. The Crane valve comprised a change valve and two distributing valves similar in construction to that of the

ordinary slide valve used with the common slide-valve steam engine, except that the distributing or cylinder valves were double.

Fig. 60, which shows a section of the valve and a portion of the cylinder face, gives a clear idea of its construction. *A* and *C* are the cylinder ports leading to each end of the cylinder. *B* and *D* were used either for exhaust or supply ports as required. The annular space at the back of the distributing valve was simply a passage for the service of port *A*. If the supply of steam came through port *D*, it passed through the passage at the back of the valve and was ready to pass into port *A* whenever the valve was moved far enough to open that port. On the other hand, if by moving the change valve the steam was admitted through port *B*, then port *D* became the exhaust port and the passage at the back of the valve would take the exhaust steam away from port *A* to *D*. Changing the direction of the steam in the manner described above would produce a reverse motion of the engine with an exactly similar motion of the distributing valve, so that but one eccentric and rod were required to produce and govern both motions. But with a reverse motion produced in this manner it would be impracticable to have either lap or lead with the valve, for the valve must be so made that it just covered the cylinder ports when central, as shown in the cut. This was the cause of a great deal of waste, but it could not be avoided, and in the case of a spur-gear engine it was necessary for safety.

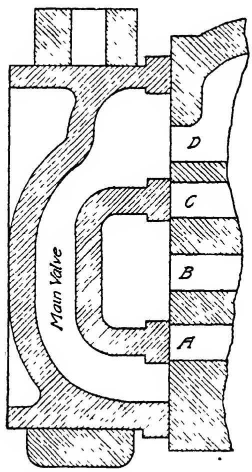


Fig. 60. Section of Crane Steam Valve

A casual inspection of these distributing valves will show that the area of the back of the valves exposed to the pressure of steam in the common steam chest provided is greater than the area of either of the ports in the valve, and only one of these ports is used at a time for line steam. Hence, because of the greater area of the back of the valves and, therefore, of the greater pressure, they are always held to their seats. However, if the elevator car should be very much overloaded, it is possible for the car to drive the engine if spur gearing is used. In such a case the engine would run back-

wards and pump air into the boiler. The pressure under the valves would become greater than that on the outside, and the valves would be forced off their seats. The car would run quickly to the ground and the valves remain off their seats, thus crippling the engine. To prevent such an occurrence, guides were bolted to the bottom of the steam chest in such a way that, while permitting the free travel of the valve, the latter was held tight against its seat. Such an arrangement was absolutely essential in an engine fitted with this type of valve motion if spur gearing was used. With worm gearing it was, of course, impossible for the car to drive the engine.

## TYPE OF ENGINES

**Vertical Cylinder.** The engines used at this time were 7-inch bore of cylinder and 9-inch stroke, and the steam pressure was from 80 to 90 pounds per square inch at the boiler. The engine's speed was up to 200 to 225 revolutions per minute and, with the gearing and drum already described, the speed of the car with the worm gearing was about 100 feet per minute, while after the introduction of the internal spur gear a car speed of 200 feet was easily attainable. This was, at that time, the greatest speed which had been attained with elevators. It soon became apparent that higher speeds could be had if the means of stopping were adequate. This feature proved to be a very serious obstacle, and one which was not overcome until later, when the lever device for use with hydraulic elevators was evolved. Many operators developed marvelous skill in handling cars at what were considered high speeds, but these were exceptions.

All the engines in use at this time were similar, being of the vertical cylinder type. For greater loads an engine having cylinders of 8-inch bore and 10-inch stroke was used, while for light loads of one ton or less, a 6- by 8-inch engine was employed.

**Direct Connected.** In the seventies an engine which was directly connected to the worm shaft was built, Fig. 61. The same floor type of worm gear and housing was used, but it was bolted to an extended bed, the worm shaft being also extended. Where the worm shaft projected beyond the housing it was formed into a double crankshaft, the bed being provided with pillow blocks for it to run in. Above this the cylinders and steam chest were set on suitable frames, with sufficient space provided between the cylinders

and crankshaft to allow room for the guides, crossheads, and a proper length of connecting rod, the valves and valve mechanism being either of the Crane or Otis type.

The initial machine, which was installed in a building on State Street, Chicago, was somewhat unsatisfactory at first, but eventu-

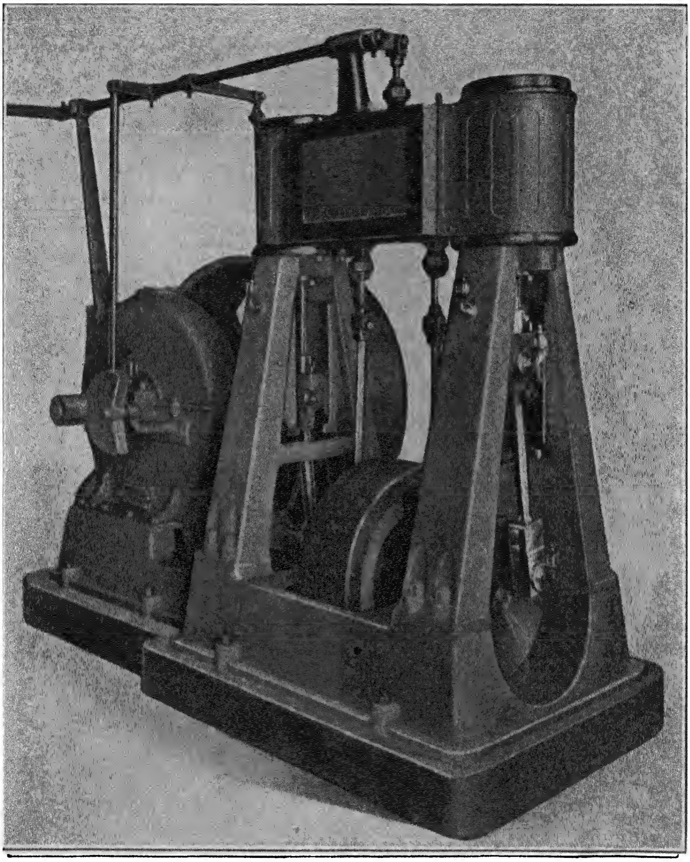


Fig. 61 Direct-Connected Steam-Driven Elevator

ally this type of machine became very successful. The principal modifications found essential to success were: double-thread worm and gear, larger diameter of winding drum, and heavy balance wheel on the crankshaft between the cranks. The heavier balance wheel produced a more steady motion in the engine, especially when running at low speeds, while the worm of greater pitch added to

the speed of the car, as did also the larger drum. The speed of the engine was also increased by making the steam passages of greater area. These engines were operated through the medium of a hand cable, because at that time it was the only known means.

**Introduction of Pilot Valve.** Later, when the pilot valve was introduced with the hydraulic elevator and the hand cable was displaced by the lever-operating device, it was seen that this medium of operation would be an improvement if applied to the steam elevator. But in order to make it available for use with this machine, a special form of change valve had to be devised. To supply this need a valve was made in the form of a hollow cylinder, which was turned to fit accurately in a valve casing previously bored true and smooth, and a ground fit.

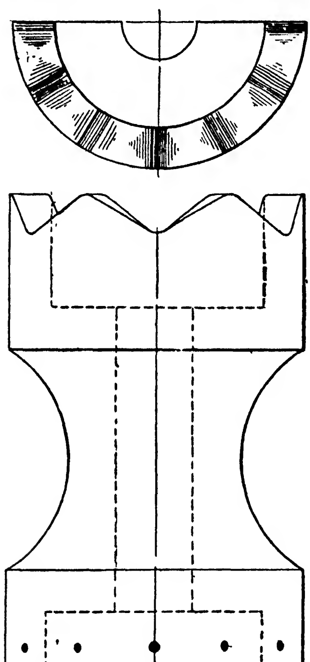


Fig. 62. Special Type of Steam Valve

The advantages of this form of valve were that very little power was required to move it because of the valve being balanced, and that the ports in the valve casing were of increased area as a result of the cylindrical form. The valve was 3 inches in diameter, so the port in the casing, which extended entirely around the valve, had a virtual length of more than 9 inches. The travel of valve, therefore, could be much less than in the flat type. In order to admit and cut off the steam gradually, that end of the valve which let in steam to the hoisting ports was made wavy, as shown in Fig. 62. For the lowering end, only a few holes were necessary, as it required but very little steam to run the engine at a fast speed while lowering. At the same time, the load was never able to drive the worm, turn the crankshaft, or move the pistons and rods without the aid of steam, hence no brake was required.

The introduction of this type of change valve, which was used only with the direct-connected worm-gear machine, permitted the

application of the lever device to the steam elevator. With this improved method of control, much higher speeds were made practicable. Then another change was made, namely, the use of the three-thread worm and gear. This made it quite easy to reach car speeds of from 350 to 500 feet per minute with the larger engines and maintain complete control. Hence there was considerable competition between the makers of steam and hydraulic elevators.

The introduction of the electric elevator with its high speeds and its economical operation and maintenance gradually eliminated the steam elevator, and today there is hardly one to be found. But the fact remains that it was in its day a very efficient machine and more durable than the electric, the electric parts of which deteriorate rapidly.

## HYDRAULIC ELEVATORS

### EARLY FORMS

**Armstrong Hydraulic Crane.** The idea from which the hydraulic elevator was eventually evolved originated with Sir William Armstrong, an eminent English engineer. He was interested in a stone quarry situated in a hilly district in Yorkshire, England. About two hundred feet above this quarry was a reservoir of water used for the supply of a neighboring town. His idea was to use this water for lifting the stone out of the pit, and for this purpose he made the hydraulic crane shown in Figs. 63 and 64. It comprised a cylinder set almost horizontal—the cylinder being bored true and smooth and fitted with a piston rod—and a crosshead which ran on guides set in line with the cylinder. The crosshead was supplied with a shaft and sheave, grooved for the reception of a chain. The extreme end of the frame farthest away from the cylinder was equipped with two similar sheaves and shafts set one above the other, as shown.

One end of the chain used for lifting the loads was attached to the crosshead; after being led around the upper sheave of the frame and around the traveling sheave on the crosshead, it was carried under the lower fixed sheave, and from there up through the port of the crane and out to the end of the jib and down, terminating in a hook for hitching on to the loads, and having attached to it a heavy weight to cause it to overhaul when descending unloaded. To

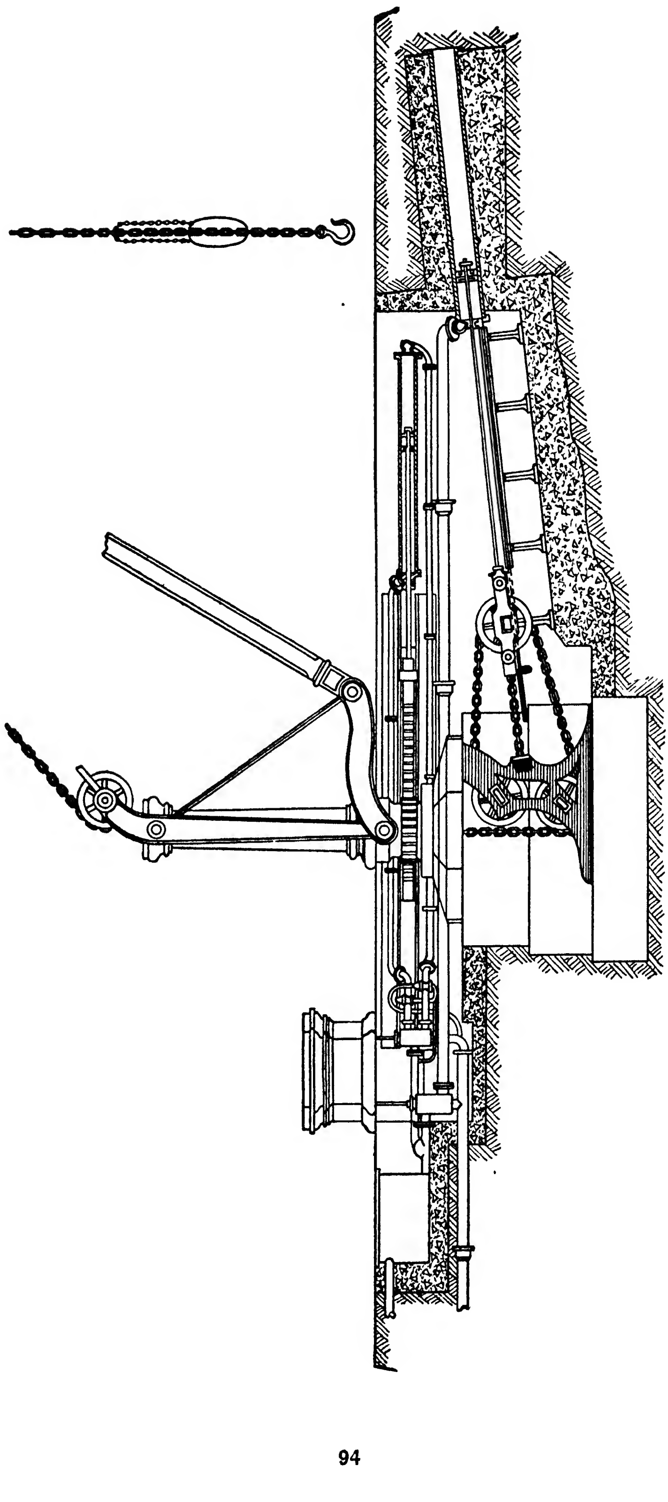


Fig. 63. Section of Armstrong Hydraulic Crane

facilitate this overhauling of the chain, the cylinder, instead of being set horizontal, was set at a considerable angle, all of which is clearly shown in the illustrations.

This machine was also equipped with an auxiliary cylinder, the piston rod of which was fitted with a rack meshing with a pinion

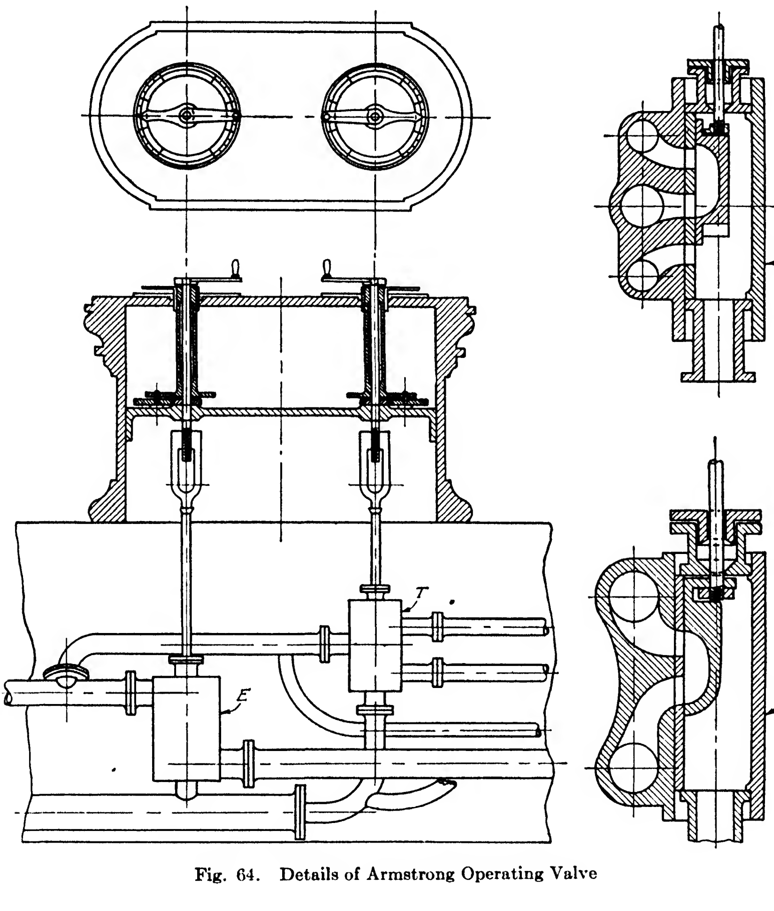



Fig. 64. Details of Armstrong Operating Valve

extending around the lower part of the crane port. This auxiliary cylinder was used for the purpose of swinging the crane around in a circle for convenience in depositing the loads at the top of the bank as well as for hitching on to them below. As this rack and piston necessarily had to operate in both directions, this cylinder had to take water at either end as might be desired, while the cylinder

which performed the hoisting took water at one end only, the other being open to the atmosphere.

*Adapted for Lifting Merchandise.* This machine was so successful for the purpose for which it was designed that a few years later it occurred to the originator that it might be used for lifting merchandise in warehouses. As a result, many such machines were made and installed in England. In some cases, in order to save room, the cylinders were set vertical. These elevators were never supplied with a car or platform traveling in guides, but always with a chain and hook. That they were not an unqualified success may be readily seen by examining the details of the operating valve shown in Fig. 64. This was of the ordinary D-type, being similar to that used as the distributing valve on an ordinary slide-valve steam engine.

## AMERICAN IMPROVEMENTS

It was about the year 1866 or 1868 that the Armstrong machine was introduced into the United States and, following the course of many other devices brought here from the Old World, it was wonderfully improved upon.

**Addition of Platform.** The first improvement made by the American engineers was the introduction of the platform or cage traveling in guides, which seemed to them so essential to a handy and convenient elevator. The horizontal type only was used at that time, and only where water under pressure was available or could be produced by pumping water to a tank on the roof. As buildings at that time seldom exceeded four to six stories in height, the pressure obtained in the latter case was never very great.

**Use of Pressure Tank.** Where a natural or artificial head was not obtainable, American engineers introduced a form of accumulator by using a pressure tank, the water of which was replenished by means of a pump. The pressure was produced by the use of air pumped into the tank under pressure, about one-third of the volume of the tank being occupied by air when the tank contained the maximum quantity of water used. The elasticity of the air permitted the elevator to make several trips before it became necessary to operate the pumps again. Where either a roof or pressure tank was employed, the same water was used over again, being discharged

into a tank below the surface of the ground and then pumped back into the roof tank for use again; but where water was obtained from the street mains, it was discharged into the sewer.

**Two-Way Operating Valve.** *Early Forms of Operating Valve.* Another improvement made in the construction of the machine was the operating valve. Besides being difficult to keep tight, the old Armstrong valve, through having the pressure all on one side,

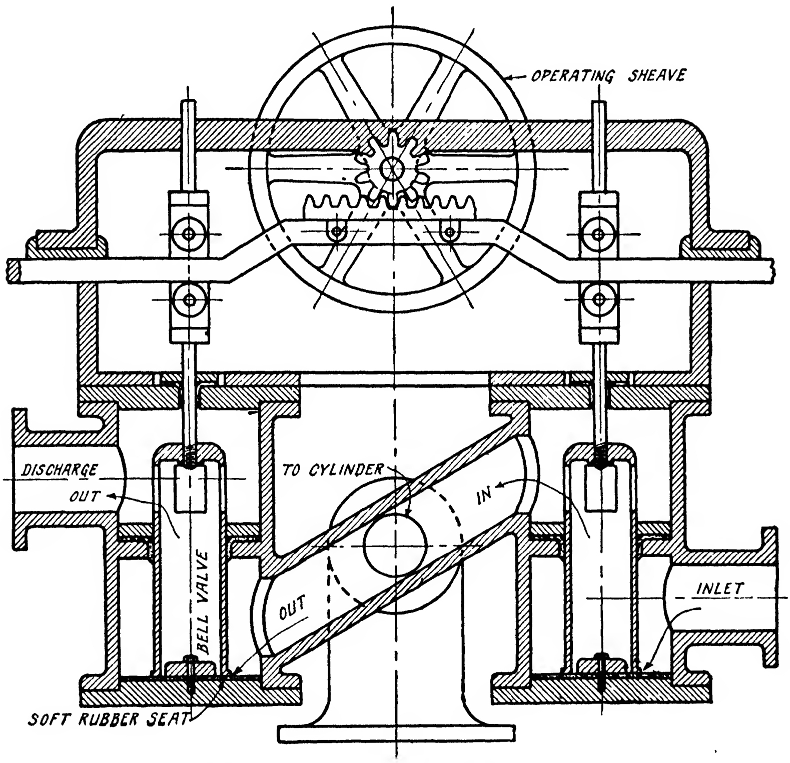



Fig. 65. Bell-Type Two-Way Operating Valve

required a great amount of power to move it. This feature made it unsuited for operation from a moving platform, so recourse was had to a design which was more nearly balanced, that is, one in which the pressure was not all confined to one side of the valve. Many types of valve were introduced, such as double poppet valves, and the so-called bell-type and bottle-type valves, Figs. 65 and 66. As all, or nearly all, of them fell into disuse and became obsolete,

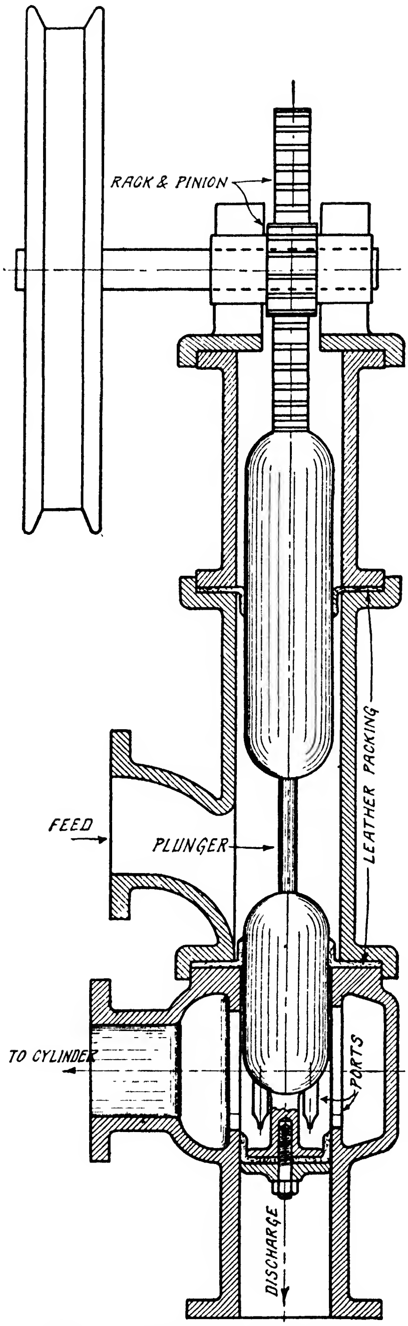



Fig. 66. Bottle-Type Two-Way Valve

only that one which most nearly filled the requirements, and which came into general use, will be described.

*General Description of Two-Way Valve.* The body of the two-way valve proper, which comprises the two middle sections *h h* shown in Fig. 67, consists of two short cast-iron tubes with flanges for bolting them together. These tubes are accurately bored and the flanges faced, and each section is lined with a piece of seamless drawn-brass tube one-eighth of an inch thick. This tube is as true and smooth as if turned and bored in the lathe, and is made a tight fit in the cast-iron casing, being forced into place under pressure. The lining *b* in the upper section is plain and does not extend the full length of the cast-iron body. It will be noticed that each section of this cast-iron body is somewhat enlarged where the nozzle or port enters. This enlargement permits of forming a passage for the water all around the brass lining to provide a full flow of water with as little hindrance as possible.

*Brass Lining.* The brass lining in the lower section extends from one-half inch above the upper flange of this section clear through to the lower flange. At

the part which comes opposite to the lower port or branch marked "To Cylinder", Fig. 67, the tube is perforated by numerous holes of various sizes set evenly in rows. The combined area of all these holes must be one-third greater than the area of the internal section of the brass tube. The upper and lower rows of holes must be smaller than those toward the middle and not so numerous, the object of this being to admit and retard the flow of the water gradually. The extreme end rows of holes are usually $\frac{3}{16}$ to $\frac{1}{4}$ inch in diameter and those toward the middle increase in size, row by row, until they attain a diameter of from $\frac{7}{16}$ to $\frac{1}{2}$ inch. The laying-out and drilling of these holes is an important feature in the making of a valve, for upon it depends the smooth and gentle starting and stopping of the elevator. The operation of making these holes is called the "graduating" of the valve. The size of the holes, their distance apart, and their number vary with the size of valve used and the pressure under which it is to work.

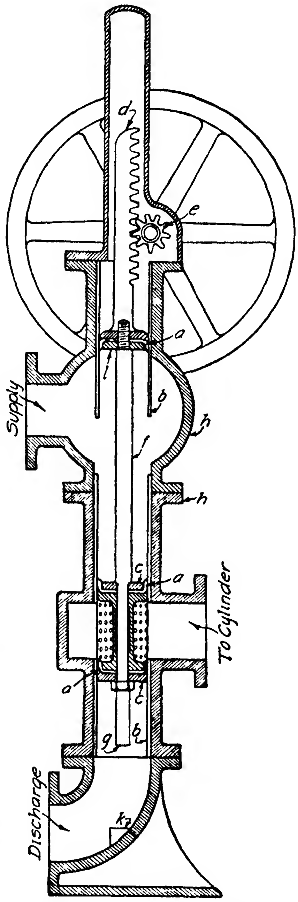


Fig. 67. Improved Type Two-Way Operating Valve

The upper end of this section of the valve lining is allowed to project one-half inch beyond the upper flange, in order that when the valve is assembled it may enter the lower portion of the upper section and thereby keep the two sections in line and true. To prevent leakage of water a "gasket" of paper, cut the size and shape of the flanges and dipped in boiled linseed oil, is slipped between them, and this is all the packing required for this joint.

On the top end of the upper section is bolted a hood, which forms a cover or protection for the upper end of the plunger and contains two boxes or bearings in which the pinion shaft *e* may revolve. At the bottom end of the lower section is bolted a casting which serves

the double purpose of forming a base for the valve to set upon and also an elbow and discharge port for the valve.

*Plunger and Packing.* The plunger *f*, of this valve comprises a stiff steel rod or shaft, which is turned smaller at its upper end and threaded. Over this is slipped a brass washer or flange *l*, the office of which is to fill out and support the leather cup *a*, which is used as a part of the packing of the plunger. After this leather cup is put in place, the rack *d* is attached. This rack, which is previously put in the lathe and turned true at its lower end and along its whole length directly back of the teeth, is then drilled for a distance of one and one-half inches and tapped out to fit or screw on the end of the valve stem, thus holding the leather cup firmly in position. The other end of the valve stem is similarly treated. It is turned down smaller, but for a greater distance, to receive another brass casting called the spool and two brass cups or graduators. These castings form holders for leather cups, which are the packing at this part of the valve, and when in place are held there firmly by a brass nut and locknut. The end of the valve stem below these nuts is turned down smaller for a certain distance as at *g*, Fig. 67, and serves as a stop by striking on the projection *k*, thus preventing the plunger traveling too far in that direction. Its travel upward is stopped by the lower portion of the rack striking the hood.

*Operation.* The action of the valve is as follows: The position of the plunger, as shown in Fig. 67, is the neutral or stop position, that is, no water can get in or out of the cylinder while the plunger remains as shown. A movement of the wheel will cause the plunger to rise or fall according to the direction in which the wheel is turned, this being accomplished through the medium of the rack and pinion, the wheel being keyed to the pinion shaft.

Should the plunger be depressed, a clear path will at once be made for water to pass from the supply opening to the cylinder. Upon returning the plunger to its present position all motion ceases and the elevator comes to rest. If the plunger is raised, a clear passage is provided for the escape of water from the cylinder to the discharge opening. Permitting water to enter the cylinder from the supply source causes the elevator to rise, while allowing it to escape produces the opposite movement of the car or cage.

There are two features of this valve which it would be well to

explain before leaving the subject, namely, the function of the upper cup and the principle and operation of the leather cups.

*Function of Upper Cup.* The upper cup is what produces the balance of the plunger. If the hood were water-tight and there were no upper leather cup, the water upon entering the valve through the supply would immediately force the plunger down to its lowest point of travel. It would then require considerable power to raise it to its neutral position. If the water pressure was 100 pounds per square inch and the valve of the 4-inch type, thereby giving a pipe area of 12.56 square inches, the pressure on the plunger holding it down to the lower end of the tube would be slightly over 1250 pounds. But by using an upper leather cup, and by screwing into a hole tapped in the lower part of the hood a discharge pipe for carrying off any leakage water, an opening above the upper leather cup to the atmosphere is provided. When the supply water enters the valve, it presses upward against the leather cup with as much force as it does downward against the lower portion of the plunger, thus balancing the pressures on the two sides of the plunger, the only resistance to its movement through the tube being the friction between the sides of the tube and the plunger.

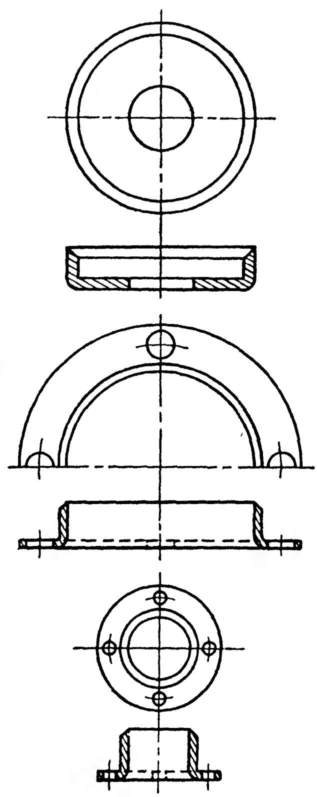


Fig. 68. Details of Leather Cup Construction

*Construction of Cups.* The leather cup is a very simple and efficient form of packing. The principle on which it works was discovered nearly a century ago by Joseph Bramah, who had experienced great difficulty in packing the ram of his hydraulic press. He finally overcame the trouble by means of a number of leather collars made as shown in Fig. 68. The leather cup used in the hydraulic valve is of the same type, but inverted, and is as nearly free from friction as anything of the kind can be made.

Belt leather $\frac{3}{16}$ of an inch wide makes the best cups because it is softer and more pliable than sole leather. After being soaked in water until soft and pliable, it is forced into a mold consisting of a collar, the base of which is of the same diameter as the outside of the cup is to be, and a smooth and true plug of the same diameter as the inside of the cup is to be. The inner edge of base of collar and the outer edge of plug are rounded so as not to damage the leather and to give the cup its proper shape. After the leather is laid on the collar in such a way as to have the hair side of the leather form the outside of the cup when made, the plug is slowly forced fully into the collar. The leather is then kept in the mold until dry, after which it is roughly trimmed by hand and put into a chuck in a wood-turning lathe and properly and truly turned, the "featheredge" being formed in this operation.

When the cup is made and put in place, it is, while in use, constantly immersed in water and would become soft and pliable again and rapidly assume its former flat shape, if not properly supported and protected. Hence the spool and other parts which hold the cups while in place in the plunger of the valve are shaped in such a manner as to retain the cup in proper form while in use.

It will be readily seen from this description and the accompanying illustrations that this packing is in a measure automatic in its action, for the pressure of the water on the inside of the cup is the cause of its holding the water. For this reason the holes in the brass lining must not be too large or the leather will be forced through them. Many makers have resorted to the expedient of milling a series of longitudinal slots in the lining instead of drilling round holes as described, but there seems to be no particular advantage in this and special machinery is required to make them. The leather cup is an important factor in the successful and efficient valve, and will be found in a more or less modified form in all that will be described hereafter.

## DEVELOPMENT OF LATER LOW-PRESSURE TYPES

**Early New England Type.** About the time the horizontal hydraulic elevator of the Armstrong type came into use in the United States, another and similar machine of great simplicity was developed in New England.

In the State of Maine at that time, many of the stores and warehouses in cities and towns were low buildings of not more than three or four stories but of large area. Instead of using cylinders of comparatively large bore and a number of multiplying sheaves, long cylinders of smaller bore were substituted, with no multiplying sheaves. These cylinders were of the same internal diameter and made in sections of 10 or 12 feet in length bolted together to form one long cylinder, the length of which was equal to the travel of the cage. They were carefully lined up on piers of masonry set in the ground beneath the first or ground floor, as shown in Fig. 69. The piston was about three or four feet in length, and about four $\frac{5}{16}$-inch or $\frac{3}{8}$-inch steel rods were used in place of the regular piston rods. These rods passed through separate stuffing boxes in the cylinder head and under a sheave of large diameter, grooved to receive them, and thence up the hatchway to a similar sheave at the top of the run, and over this sheave down to the cage as shown.

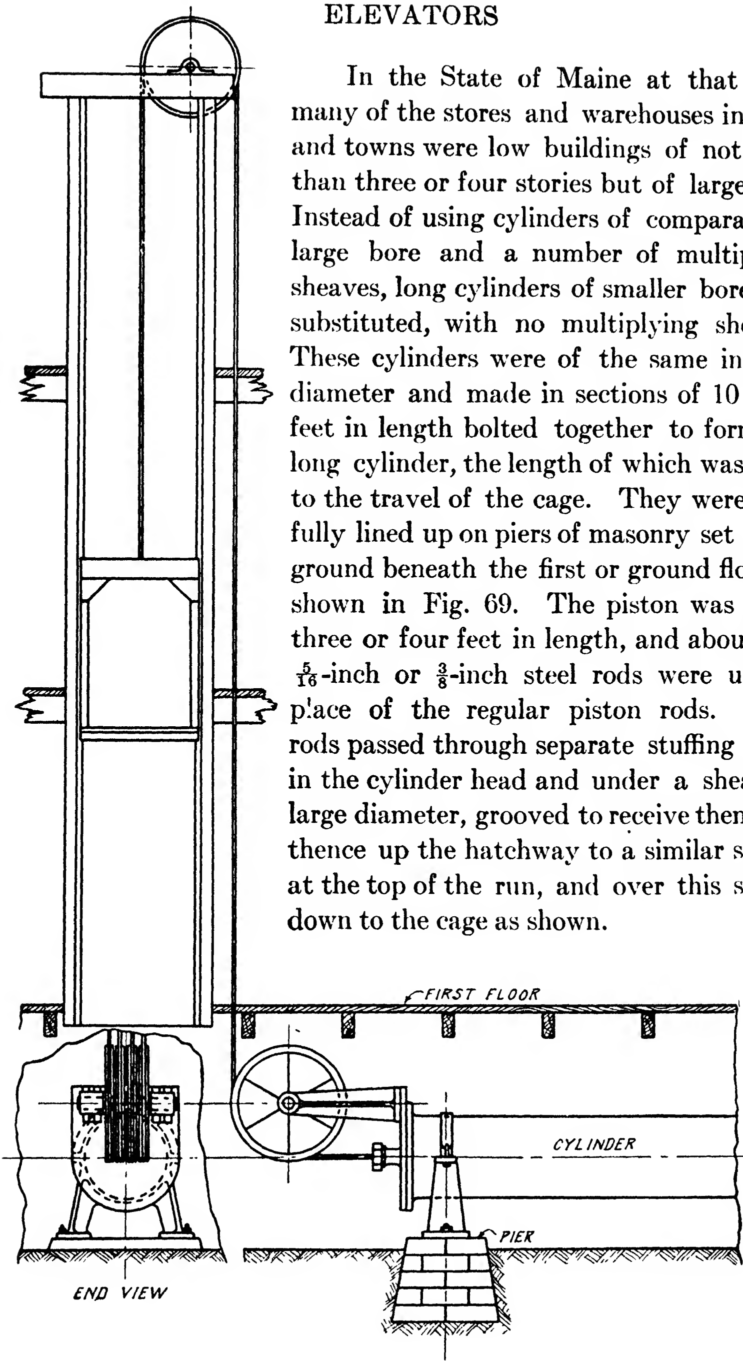



**Fig. 69. Early New England Type of Low-Pressure Hydraulic Elevator**

These rods, or wires, were moderately flexible and would readily bend around a 4-foot sheave, without danger of becoming crystallized and cracking for some time, and enough of them were used to insure the necessary tensile strength. No counterweight was used to offset the weight of the car, as it was believed that all the weight of the empty cage was necessary to overhaul the long heavy piston. The valve was the one which has just been described.

These machines were not available as vertical engines on account of the extreme length of cylinder required and because the available water pressure would not always carry the supply as high as the top of the cylinder. In practice, these machines were a disappointment. It was found that, owing to their diminutive size and the speed at which they traveled, the piston rods cut not only the packing but also the stuffing boxes very rapidly. It was also impossible to lubricate either the pistons or the cylinder, which were inaccessible, and as a result they became leaky and wore out quickly. The steel wires crystallized and cracked much sooner and more often than was anticipated. Passing notice is given this machine only because of its novelty and the experience it afforded.

**Hale Water-Balance Elevator.** About the year 1869 or 1870 Mr. William E. Hale patented and introduced a machine, known as Hale's Water-Balance Elevator, which in many particulars greatly resembled the one just described. The novelty of this machine was that it utilized the force of gravity for its operation.

*Construction.* At the top of the run, one or more sheaves mounted in the customary manner spanned the distance from the center of the hatchway to the center of a large tube about 3 feet or more in diameter, as shown in Fig. 70. This tube extended from this point to the ground, being run as closely adjacent to the hatchway as possible. The lower end of this tube terminated in, or connected with, an underground tank. On the roof directly above this tank were another tank and a steam pump which pumped water from the lower to the upper tank through suitable piping. Over the sheaves at the top of the run was the usual cable, to one end of which was attached the traveling car and to the other end a large metallic bucket.

*Brake.* The cage, in addition to the usual guideways, was supplied with a powerful brake actuated by strong steel springs

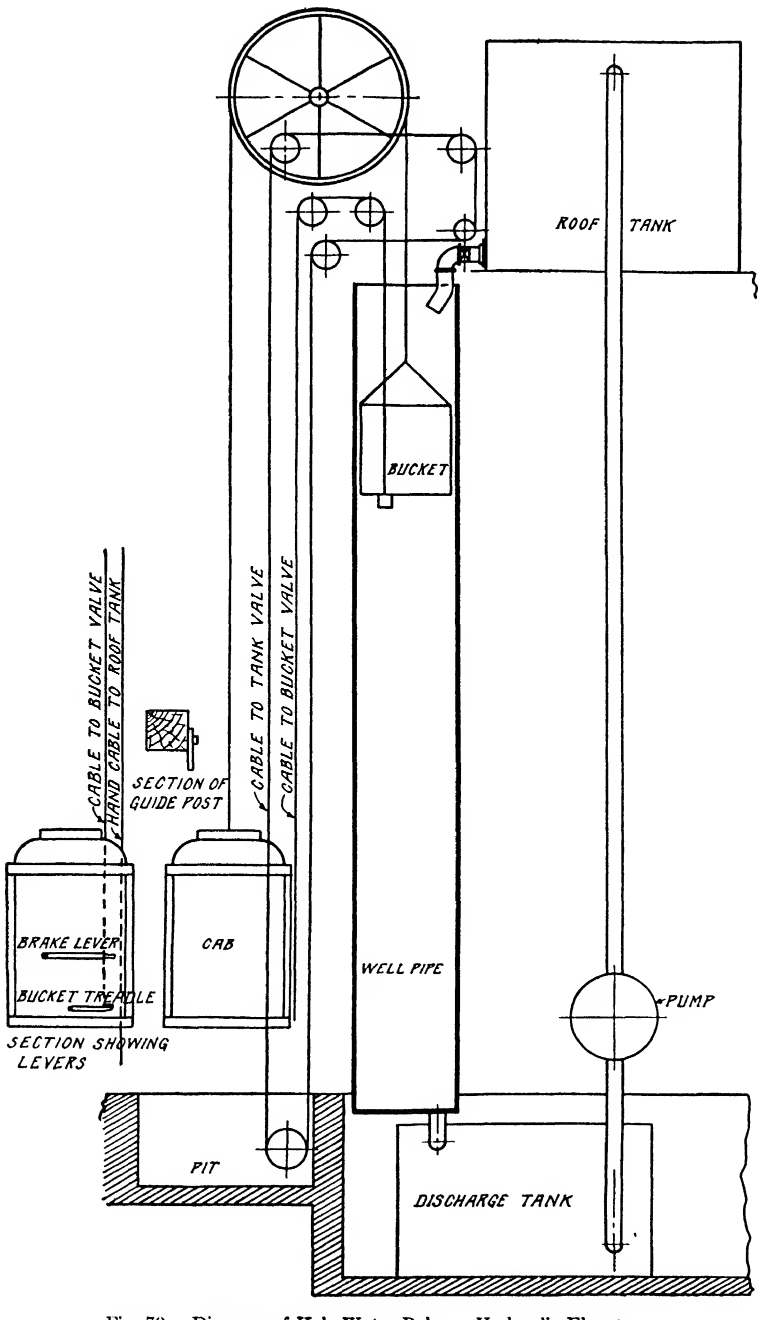



Fig. 70. Diagram of Hale Water-Balance Hydraulic Elevator

which, when left to themselves, automatically applied the brake to the guides. The guides were long, thin, flat bars of steel about $\frac{1}{2}$ inch by 6 inches, bolted to one side of the guide post, which was of wood, the steel plate projecting about 3 inches beyond the post to form a guide for the car and to present a suitable surface for the long brake shoes to act upon. This brake also served as a safety device. It was released with considerable effort by means of a lever located inside the car.

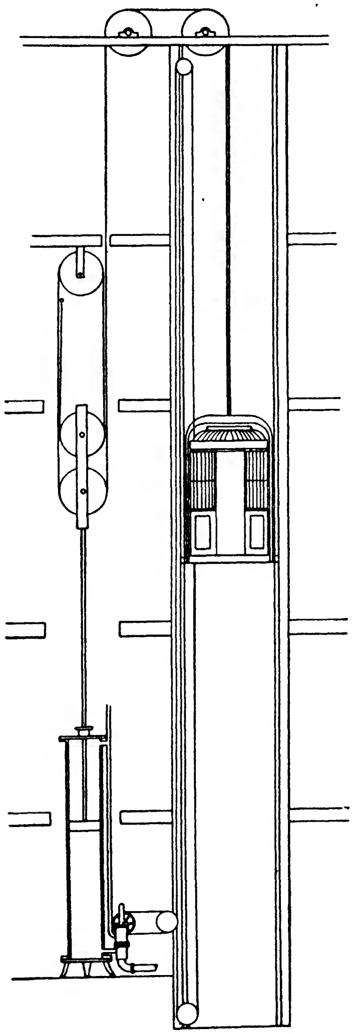


Fig. 71. Diagram of Hale Standard Hydraulic Elevator

*Operation.* At the bottom of the roof tank and directly above the metallic bucket there was a large valve which was actuated from the cab by means of a hand cable. In the bottom of the bucket was a similar valve with which communication was had by means of a foot lever in the cab. When it was desired to cause the elevator to travel upward, the brake was released and the valve in the upper tank was opened. Water was permitted to flow into the bucket until its weight caused it to descend, pulling the cab upward. The stops at the intermediate floors were accomplished by means of the brake.

The descent was accomplished in a similar manner, the water being let out of the bucket into the well from whence it flowed into the underground tank, to be pumped to the roof tank again.

*Speed.* The speed of this elevator was remarkable. It exceeded that of any elevator that had been made up to that time. Its lifting capacity was limited only by the weight of water the bucket would hold and the strength of the materials entering into

the composition of the machine. If the guides were accurately lined up, its operation was smooth. It required skill and experience, however, to operate it and the cost of installation was also high. These were doubtless some of the causes contributory to its eventual abandonment.

**Hale Standard Hydraulic.** Another vertical hydraulic elevator manufactured by William E. Hale, and introduced three or four years after the advent of the water-balance type, was what he afterward designated as his "Standard Hydraulic". It was simply a vertical hydraulic elevator, Fig. 71, similar to the Armstrong elevator when made with the cylinder vertical, except that it used a circulating pipe which permitted the water, after it had been used above the piston, to be returned during the succeeding stroke to the under side of the piston. This feature, which was found in practice to be essential to its successful operation, was originally introduced for a different purpose.

**Efforts to Obtain Economy of Water.** Many makers of hydraulic elevators had for years been trying to devise means for economizing in the use of water for operating these machines. High pressures had not been tried to any great extent, the efforts toward this end being principally directed toward using the water discharged from the cylinders for other purposes, principally that of storing it in tanks from which it could be drawn for use in lavatories, etc.

*Armstrong Three-Cylinder Machine.* Armstrong had made a three-cylinder machine, the piston of each cylinder being attached to the same crosshead. For light loads he admitted water to one cylinder only—the central one; for medium loads to the two outside cylinders only; and for heavy loads to all three. The result was no marked success, the valve motion being too complicated, and the friction of the three pistons detracting greatly from the efficiency of the machine.

*Fensom Balancing Device.* John Fensom of Toronto, Canada, had developed the scheme of using a circulating pipe connecting the front and back ends of a horizontal machine in such a way that the water, after being used for hoisting, was discharged on the lowering trip into the other end of the cylinder, and then on the next up-trip was forced into overhead tanks for use about the building.

To take care of the heavy loads, he set the bearing boxes of his overhead sheave on the end of a lever pivoted on knife-edges similar to those used with the levers of large platform scales. The long end of this lever was weighted to suit the loads as shown in Fig. 72. When a heavy load was placed on the car or platform of the elevator, it would lift the long end of the lever, having overbalanced it and the weight attached. The long end of this lever had connected to it a small cable or a thin rod running down to a pipe laid underground and connected to the sewer. This rod or cable was attached to a valve set in the pipe and when, through an overweight on the platform, the lever was raised, it opened the valve in the pipe and allowed the discharge water to run to waste, thus relieving the

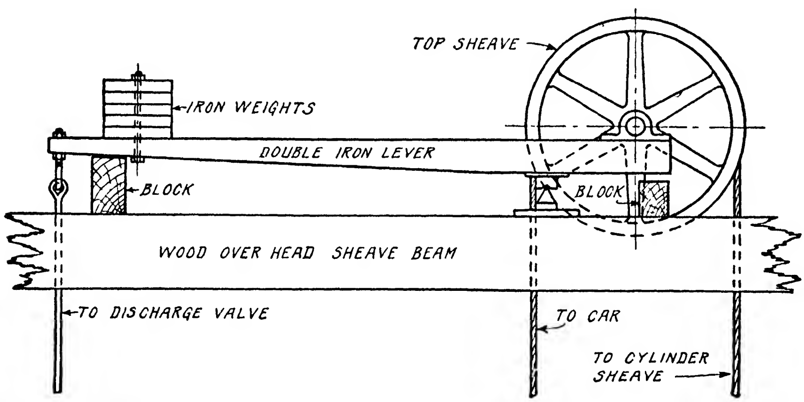



Fig. 72. Fensom Balancing Device

piston of back pressure and allowing the elevator to lift the heavy load. Hence it was only while lifting light and medium loads that water was economized.

It was found in practice, however, that the discharge water from these machines, owing to the necessarily frequent use of lubricating oils inside the cylinders, was too foul and greasy to be suitable for any other purpose and the idea was abandoned.

*Baldwin's Efforts.* The inventor of the Hale standard hydraulic, Mr. Cyrus W. Baldwin, had in view the idea of water economy when he invented that machine. He, however, labored under greater disadvantages than did Mr. Fensom. He soon found that his scheme of economizing water was not only a failure but that, owing to

universally lower pressures in the territory in which he operated, he was unable to force the discharge water into reservoirs and that in order to obtain a uniform piston pressure throughout the entire stroke he had to retain the circulating pipe and return the water to the under side of the piston to enable him to maintain this uniform pressure. Moreover, a perfect circulation of water could not be insured if his cylinders exceeded 32 feet in height, owing to the fact that a column of water 33.94 feet exerts a pressure equal to one atmosphere or 14.7 pounds per square inch.

**Hale Valve.** The Hale operating valve, Fig. 73, was identical in design and construction with that used with the other types of elevators, but it was connected differently. In the Hale standard hydraulic elevator, the piping was so arranged that the pressure of the feed water was always on the piston. The hoisting was done by opening the discharge end of the valve, Fig. 74, and allowing the water below the piston to flow away to the sewer or discharge tank, as the case might be. As the water beneath the piston flowed away, the piston would follow it, having the pressure of the feed water above it and the weight of the column of water below, both forcing it in the same direction. In lowering a load, communication was opened between the upper and lower ends of the cylinder. The pressure was then equal on both sides of the piston and the car descended by gravity. In doing so it displaced the water in the cylinder from the top to the bottom side of the piston through the medium of the circulating pipe.

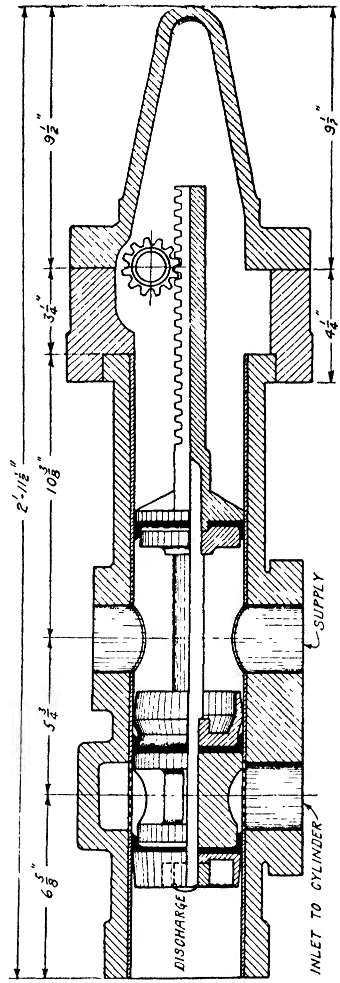


Fig. 73. Section of Hale Operating Valve

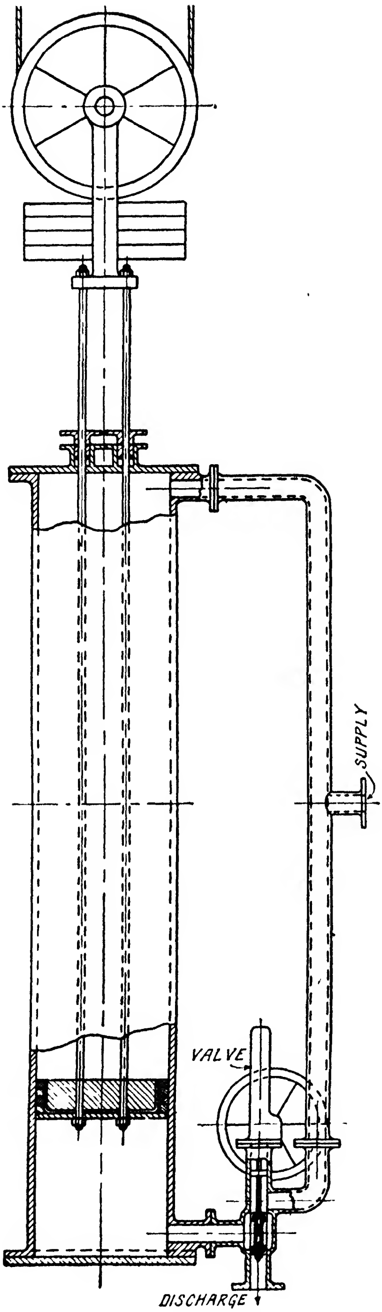


Fig. 74. Diagram Showing Operation of Hale Elevator

**Piston Packing.** The packing for the pistons of all these earlier elevators was similar to that used in the valve, i.e., a leather cup, but was much larger. So large were cups made that an entire hide of leather was often required to make one.

The comparatively rapid motion of the piston and the distance it traveled in a given time, especially in those elevators with a gear ratio of only 2 or 3, wore out this packing very quickly. The frequent renewals required and their tedious and expensive nature soon caused the introduction of other forms of packing. In some cases several rings of plaited hemp were used, the piston being turned to receive them. They were kept in place and tightened up when leaky by a follower ring held in place by bolts or by studs and nuts. In other cases, round and square cords of India rubber, set in alternately, or alternate rings of round rubber cord and square plaited hemp were substituted for the hemp packing first mentioned, each type having its advocates.

*Hale's Hemp Packing.* One of the best of these piston packings was that devised for the Hale elevator. This was a combination of the leather cup and the square hemp or canvas packing, Fig. 75, the latter being so arranged that it rubbed

against the bore of the cylinder and took all the wear. The leather cup, being disposed inside these rings and receiving the pressure of the water on its inside, forced the packing out of place but was not of itself subjected to any wear. This packing required no setting up and was automatic in its action until the rows of canvas packing were worn beyond usefulness.

**Siphon Relief.** *Necessity for Counterbalancing Adjustment.* At this time the counterbalancing of elevators had not received the attention that it has since. It was impossible to counterpoise these machines except from the cage or car, and then not up to the entire weight of the car because a certain preponderance on the car side had to be allowed in order to enable it to overhaul the lifting cables and move the piston so as to descend when empty.

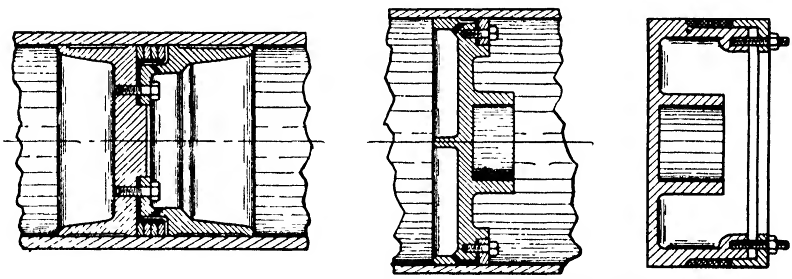


Fig. 75 Section of Piston, Showing Hale Hemp Packing in Place

In addition to this the safeguards around hatchways, except in the case of passenger elevators, were not what they are today. Most freight hatchways were open, with only a simple bar or two around them at a height of two to four feet to prevent persons from walking into them. For this reason, it frequently happened that goods were either piled on the floors or inadvertently left projecting slightly into the hatchway so there was danger that they might catch and block the car in its descent. When this occurred with the Hale machine, nothing further happened, and when the obstruction was removed the car simply continued its descent exactly as it was doing when interrupted. A glance at the illustration of the Hale cylinder and circulating pipe, Fig. 74, will make this clear.

With the horizontal cylinder it was different. Upon the car being stopped by an obstruction in its descent, all the water in the

cylinder would run out into the sewer or discharge tank. When the obstruction was removed from below the car, unless water had first been re-admitted to the cylinder sufficiently to lift it, the car would drop to the bottom of the runway, causing damage to the cables, sheaves, crosshead, or piston and cylinder head, or to the car itself, if no other casualty occurred.

*Construction.* To prevent this the "siphon relief" shown in Fig. 76 was introduced. This was simply a siphon set in the dis-

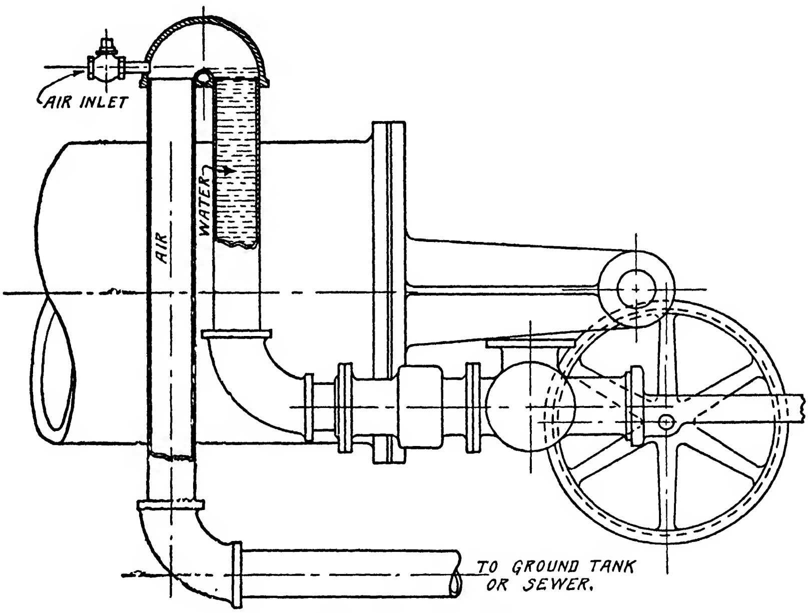



Fig. 76. Diagram of "Siphon Relief" Device

charge pipe between the valve and the sewer or tank, the upper end or bend of which was above the top side of the cylinder. In this upper bend a small check valve opening inward was inserted, and when any obstruction of the car relieved the water in the cylinder of the pressure which was forcing it out, the admission of air through the check valve, which at once occurred, would allow the water in that leg of the siphon connecting with the sewer or underground tank to flow away, while the water remaining in the other leg would balance what was left and prevent any further outflow.

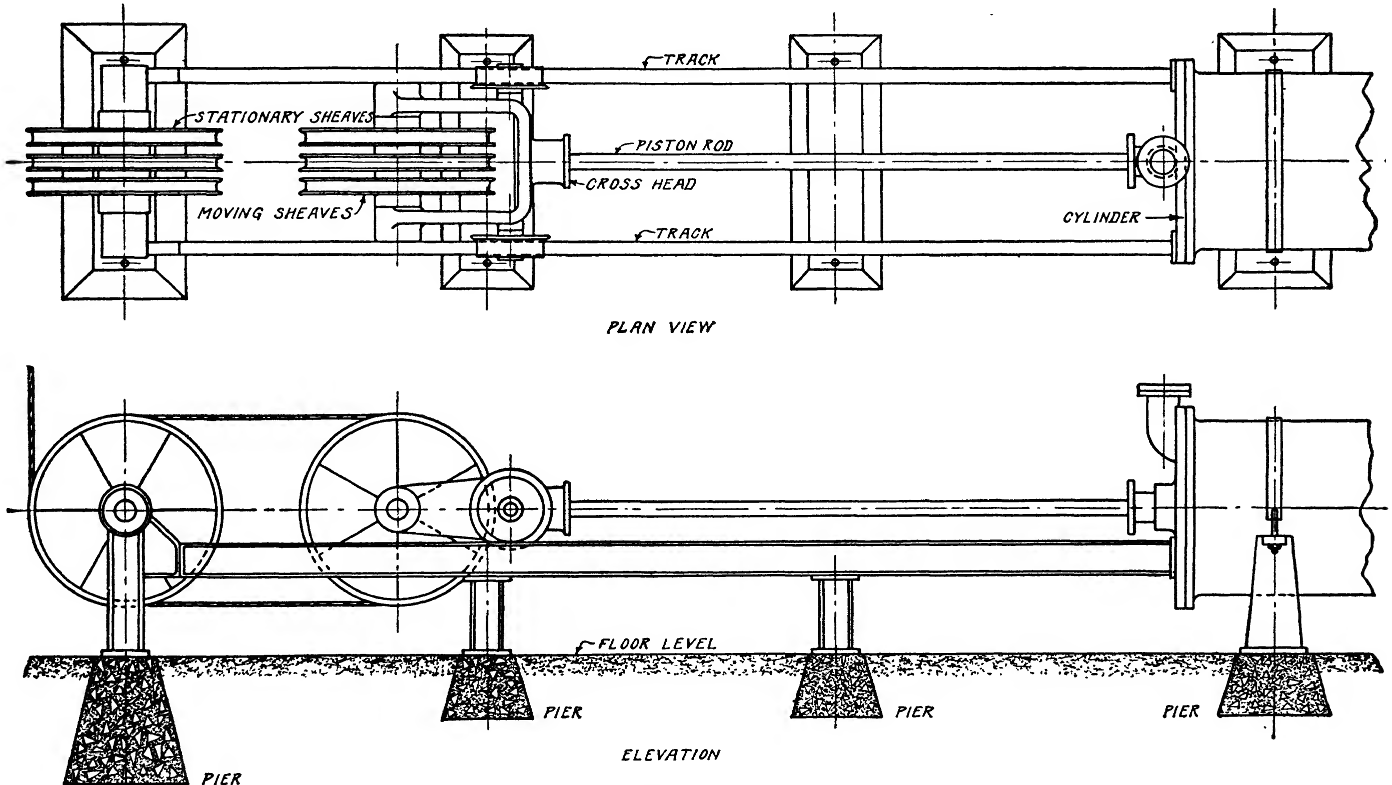



Fig. 77. Horizontal Cylinder "Pull-Type" Hydraulic Elevator

**Pull Machines.** The first hydraulic elevators made in the United States were all so built that the two sets of sheaves were pulled apart in order to produce the movement which caused the elevation of the car, Fig. 77. This form of construction had its disadvantages, the most conspicuous of which was the closeness of the two groups of sheaves when the car was at its lowest point in the hatchway.

*Pulley and Cable Arrangement.* The arrangement of the sheaves and cables on any one of these elevators, whether it be vertical or horizontal, is identical with the arrangement of the sheaves and rope in a tackle used in hoisting heavy loads by hand except as to the point of application of the power and of the load. When pulley blocks and a rope move a heavy load by hand the power available is limited—in fact, the object of the machine is to allow a small power to lift a heavy weight. Consequently, the load is applied to one end of the tackle, which is so arranged that a mechanical advantage is obtained. If four frictionless sheaves are used, a man can lift a weight four times as great as the force he exerts. However, the weight will be lifted through only one-fourth of the distance through which the free end of the rope is moved. In practice this is not strictly true, for friction has to be subtracted from the theoretical weight which it is possible for the system to lift.

With the hydraulic elevator the conditions are reversed. In this case there is an abundance of power. By applying this power directly to the sheaves and attaching the load to be lifted to the free end of the tackle, the large amount of power is so distributed as to lift a smaller load through a correspondingly greater distance.

Two sets of sheaves are used as in a rope-tackle action. One set is fastened to fixed supports while the other set is located in a crosshead attached to the end of the piston rod, as shown in Fig. 78. This crosshead is fitted with a pair of small flanged truck wheels which run on a track comprised of two iron rails set in line with the bore of the cylinder and at one end of it.

When water under pressure is admitted to the cylinder it forces the piston along its bore and, through the medium of the piston rod to which it is attached and the crosshead, it pulls the set of movable sheaves away from the set of fixed sheaves. The hoisting cable is

attached at one end to the frame which holds the fixed sheaves and is wound around one of the sheaves in the crosshead. It is then passed underneath to one of the fixed sheaves, then around, over, and back to the second sheave in the crosshead, the process being continued with all the sheaves, the cable finally being carried up the hatchway to the sheave at the top of the run, over it, and down to the car. When the piston moves, the car moves a proportionately

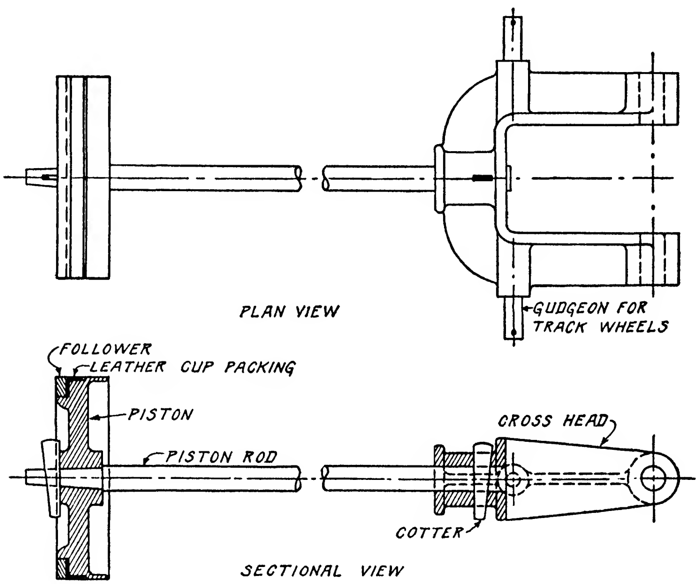



Fig. 78. Details of Piston and Crosshead for Pull Machine

greater distance, depending upon the number of sheaves that are used.

*Multiple Sheaves.* If eight sheaves are used, four fixed and four movable, the car will move through eight feet while the piston moves one foot. This is called the "multiple" of the machine. Similarly, the car will lift only one-eighth the load corresponding to the pressure applied to the piston. This, however, is only theoretical, as frictional losses will somewhat reduce the load the car can lift. Suppose the piston is 24 inches in diameter and has an area of 452 inches. With a water pressure of 50 pounds per square inch, this would give a pressure on the piston of 22,600 pounds. Let the friction in this machine be assumed to be 25 per cent. Deducting

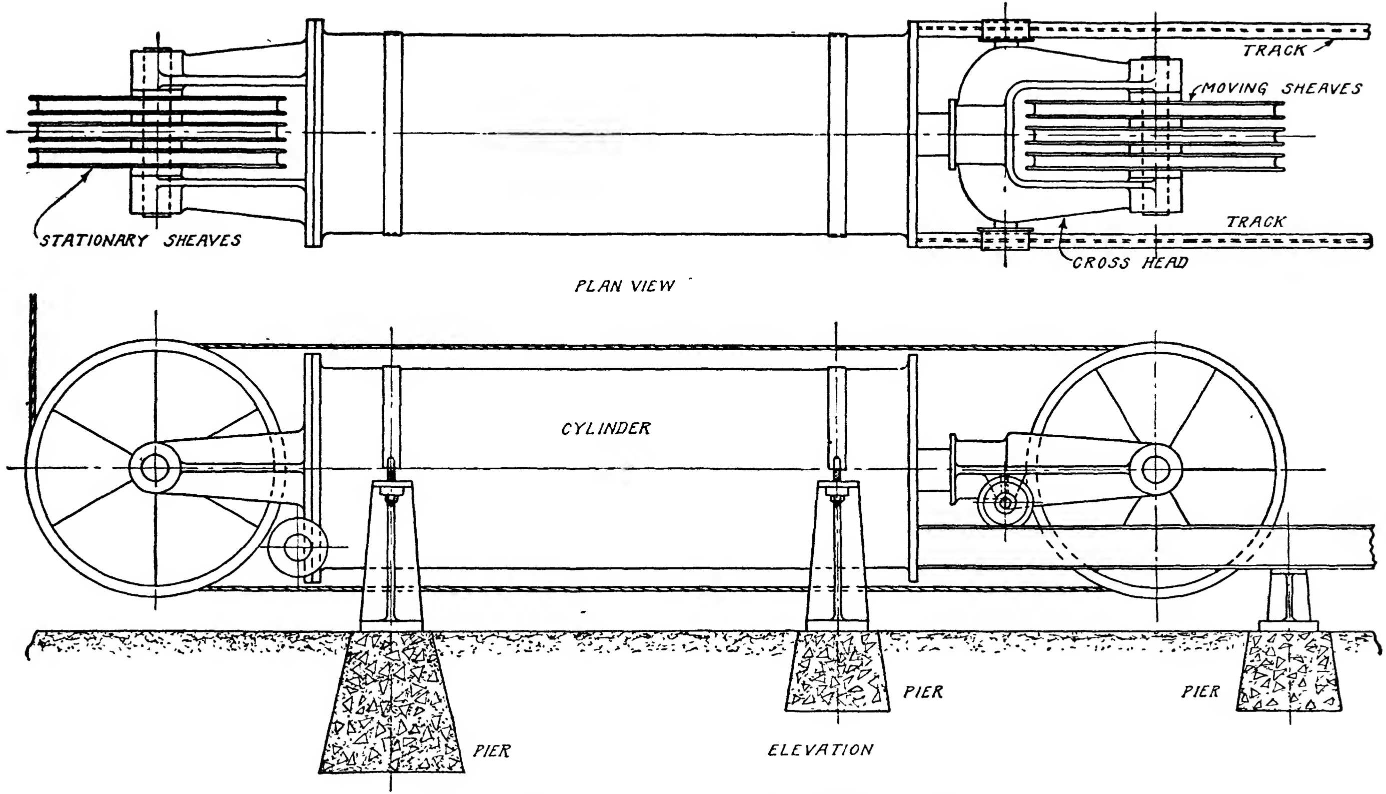



Fig. 79. Horizontal Cylinder "Push-Type" Hydraulic Elevator

the frictional loss leaves 16,950 pounds as the effective pressure. This pressure divided by eight equals 2118 pounds, the load which the machine will lift at the free end of the cable. Let it be assumed that the car weighs 1400 pounds and that it is counterbalanced to within 500 pounds of its weight, this 500 pounds of unbalanced weight being left for the purpose of enabling it to overhaul the cable and move the piston in descending. This 500 pounds must be deducted from the 2118 pounds, leaving 1600 pounds as the load which can be lifted in the car. This compared with the actual pressure on the piston seems small indeed, but it must be borne in mind that the car moves eight times as far as the piston does. While the piston is moving, say, ten feet the car moves eighty feet.

*Disadvantage of Pull Machines.* Owing to the construction of the pull machine, the two sets of sheaves were necessarily very close together when the car was at the lowest point of its travel, and as the cable passed from one sheave to the other it did not lie parallel with the center line of the bore of the cylinder but at a considerable angle. This feature caused a certain amount of friction and, in addition, chafed the cable when it entered and left the grooves of the sheaves. Of course, when the two sets of sheaves were farther apart, the angle became less acute and the consequent wear and friction was lessened.

**Push Machine.** To remedy this defect, the "push machine", Fig. 79, was devised. In this machine the set of fixed sheaves was attached directly to the cylinder head, which was moved up closer to the hatch to allow these sheaves to lead up in the hatchway as before. The piston rod was attached to the opposite side of the piston and passed out through the open end of the cylinder, thereby obviating the need of a stuffing box and eliminating another source of friction. The piston rod was made thicker, Fig. 80, because it was now to have a compressive strain instead of a tensile strain as previously. The track on which the crosshead was to travel was also set to lead away from the open end of the cylinder, as can be seen in Fig. 79. With this arrangement the two sets of sheaves never got closer together than the length of the cylinder and hence the acute angle for the cables was eliminated. This machine eventually became the most popular of the horizontal type,

though in both the horizontal and the vertical types it had serious defects.

*Use of Multiple Cables.* If the cable of the vertical or Hale machine broke, the heavy crosshead with its moving sheaves would bend over and buckle the thin long piston rods. If the stuffing boxes at the top of the cylinder began to leak badly, or if the circulation pipe broke, the lower stories of the building would be flooded with water. To prevent the damage done by the breaking of the hoisting cables, from two to six cables were used, it being supposed that not more than one, or at the most two, of these cables would break at once.

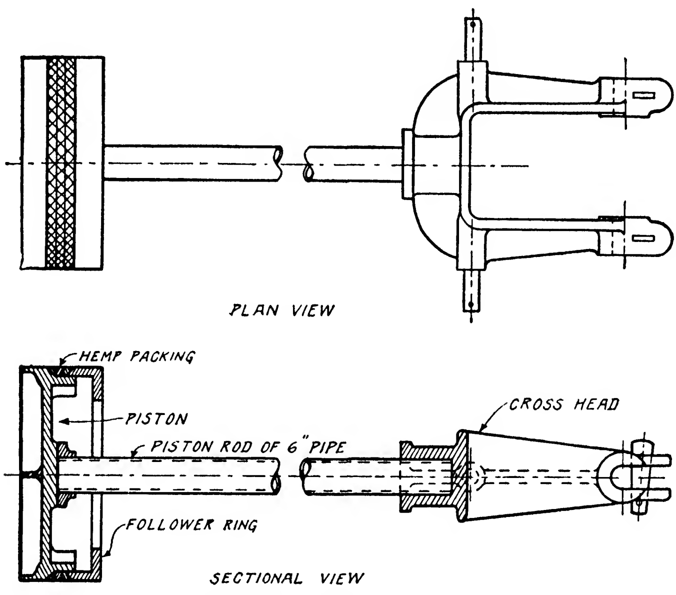



Fig. 80. Details of Piston and Crosshead of Push Machine

This feature was recognized as a safety device, and two hoisting cables instead of one were adopted on all types of drum elevators. Not more than two cables were used on the drum or winding machines because of the immense length of drum required, but on the horizontal as well as on the vertical hydraulic machine, four and even six cables for passenger machines became customary. It was found that an excessive number of cables on a hydraulic elevator increased the total friction sometimes as much as 45 and even 50 per cent of the input of power.

## AUXILIARY DEVICES

Up to this time the only method of limiting the travel of the car or, in other words, of stopping it automatically at the end of its travel at the top and bottom landings, was by means of the operating cable.

**Stop Buttons.** On this cable, at the proper distances from the lower and upper landings, were clamped balls called "stop buttons", Fig. 81. These buttons were made in halves, which were joined or attached to each other by bolts used for clamping them to the cable. An arm, one end of which was attached to the car and the other terminating in an eye, was made to travel up and down on the operating cable the full length of the run. When this eye, in the course of its travel, came in contact with the stop button, it would move the cable in the direction in which the car happened to be going, thereby closing the valve and stopping the machine.

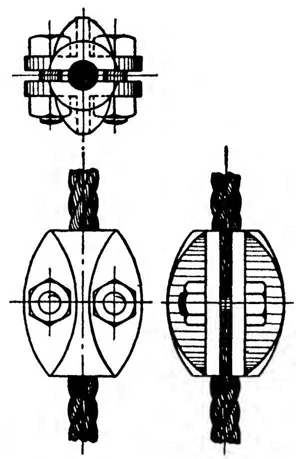


Fig 81. Details of Stop Buttons

*Protection Not Adequate.* But if the operating cable should break, through wear or through having been exposed to water at the bottom of the pit, there was nothing to prevent the piston continuing its travel until it passed entirely out of the cylinder. Hence the cellar would be flooded, for the valve being still open, a stream of water would continue to flow out through the open end of the cylinder until the valve was closed by hand. The same thing would occur if the stop button slipped, as it sometimes did. It would also happen if the hoist cables broke or became displaced from their respective sheaves, as sometimes happened with high-speed elevators when stopped suddenly.

Another disagreeable feature, in case the operating cable or valve mechanism became deranged, was that if the top of the car hit the upper beams, which carried the top sheaves, before the piston left the cylinder, the constant and terrific pull which the cables would receive was apt to strain or rupture the fastenings.

These defects were confined to the horizontal machines, and did not exist in the Hale vertical machine, because both ends of the

cylinder were closed. If the operation of the Hale machine became deranged, the piston would stop on reaching the cylinder head and all strains due to the pressure of the water against the piston were confined to the cylinder itself and no harm was done. These troubles with the horizontal machine led to the use of the *cable guard* and the *limit valve.*

**Cable Guard.** Fig. 82 shows the cable guard, which was simply a frame for holding three rods at right angles to each other close to the faces of the sheaves, so as to make it impossible for the cables to escape from their respective grooves.

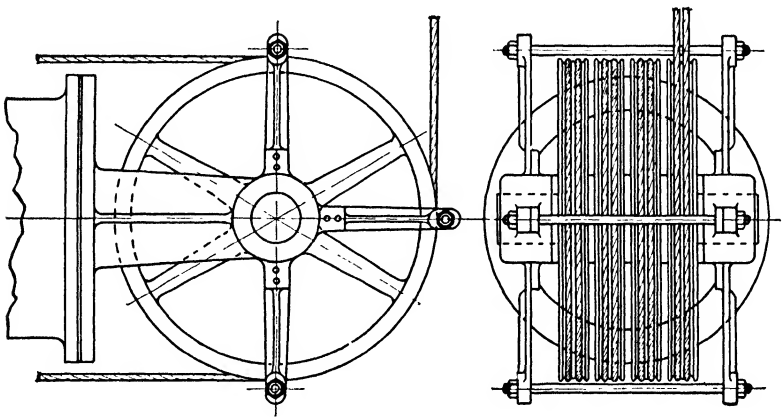


Fig. 82. Cable Guards to Prevent Slipping of Cables from Grooves

## LIMIT VALVES

The limit valve was made auxiliary to the operating valve and was placed between it and the cylinder, Fig. 83, being operated by the motion of the piston rod. The use of the limit valve necessitated a slight change in the addition of another port in the operating valve for discharging water, and an extra leather cup to control this port.

**Early Types.** Various forms of limit valves were devised. One of the earlier types for a pull machine and one that was in general use is shown in Fig. 84. In this illustration it is in the neutral position; that is, the passage is open for running in either direction, the piston being about midway in its stroke. When the piston arrived at the end of its travel in either direction, it would, through appropriate intervening mechanism, close the passage for water in that direction, thus

stopping the piston and making it impossible for it to receive any more water, regardless of how the operating valve might be manipulated, but the way was left open for the passage of water to allow the piston to move the opposite way. The methods of bringing this about were numerous; some were very simple, while others were more or less complicated.

*Operating Mechanism, Pull Machine.* One of the simplest was that in use with the pull machine. In this case, the valve stem was lengthened to a point slightly beyond the end of the crosshead guide rails. To the crosshead was attached a strong arm with

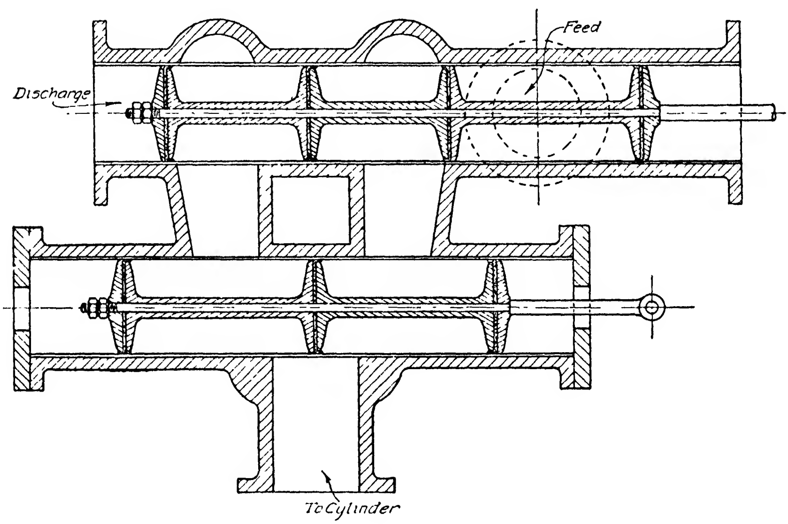



Fig. 83. Typical Section of Limit Valve

an eye which traveled over the extended valve stem in a manner similar to the way the striker traveled over the operating cable of an elevator. On this extended valve stem were collars which, by means of set screws, could be adjusted to any position desired. Being set and fastened in their proper position, when the arm attached to the crosshead reached either collar it would move the valve stem slowly along with it. This would close the valve and cut off the supply of water, consequently stopping the piston movement, but leaving the way wide open for it to go in the other direction. Spiral springs were provided which were so adjusted as to always

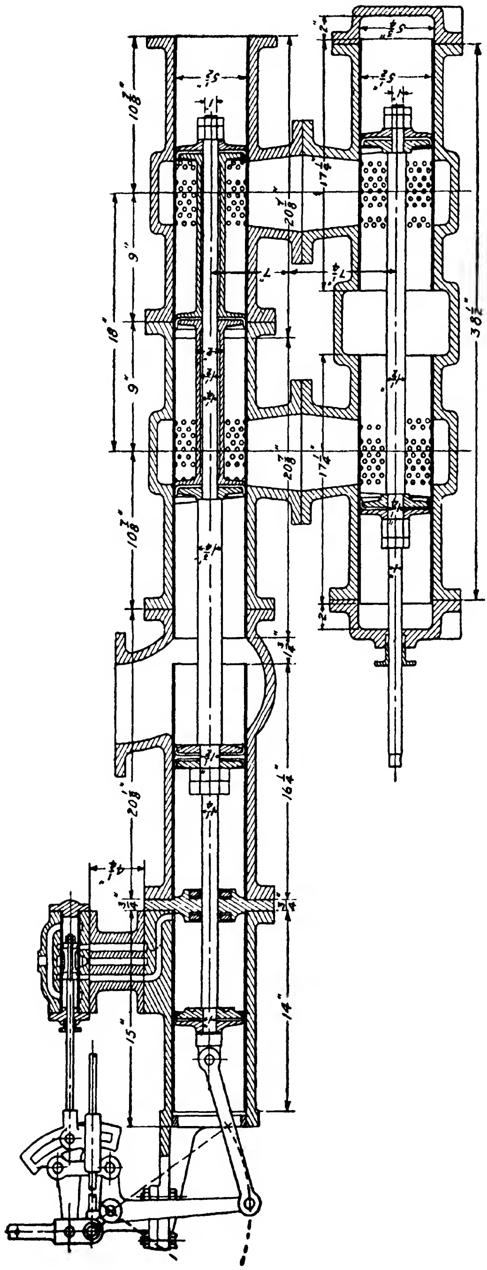


Fig. 84. Early Type of Limit Valve for Pull Machine

return the valve to the neutral or middle position, when the piston was moved in the opposite direction.

*Operating Mechanism, Push Machine.* Fig. 85 shows another method of operating the limit valve applicable only to the push machine. It comprised a channel or grooved way *A*, made with clamps *C* for its attachment to the piston rod and having its ends turned up and down, respectively, at an angle of 45°. In this groove ran a roller *D* on a pin fastened at the end of a vibrating arm keyed on a rockshaft, which worked in a bearing *B* bolted to the end of the

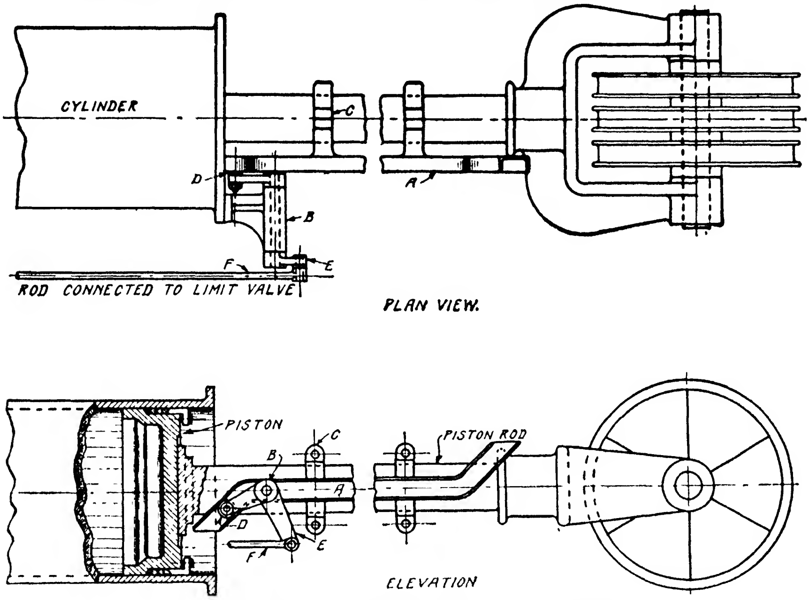



Fig. 85. Operating Mechanism for Push Machine

cylinder. At the other end of this rockshaft was another vibrating arm *E*, to which was attached the rod *F*, which opened and closed the limit valve. When the piston was in mid-stroke, the roller would be in the straight part of the channel *A*, and the limit valve would be either in the central position or open both ways, so that the engine would be free to run in either direction; but when the piston reached one end of its stroke, the arm would be deflected up, and down when it reached the other end, thus causing the valve to be closed for either the up or down motion as required.

**Whittier Limit Valve.** Another form of limit valve was used with the Whittier pull machine and was simply a form of stopcock inserted

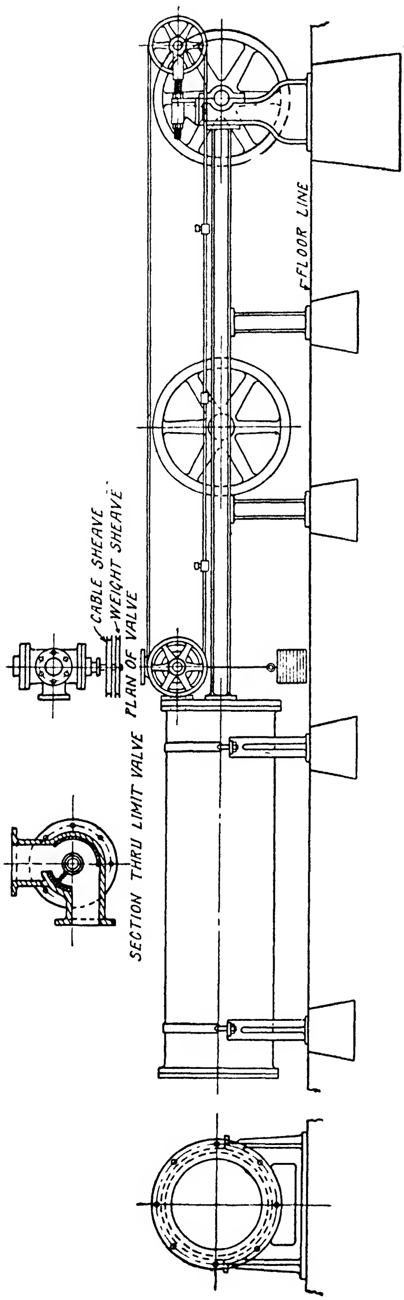



Fig. 86. Hydraulic Pull Machine Fitted with Whittier Limit Valve

between the operating valve and the cylinder. It did not close the pipe completely, but simply choked off the supply of water, allowing the piston to move very slowly at the end of the stroke until it was stopped by an obstruction bolted to the open end of the cylinder at one end or by the cylinder head at the other limit.

*Operating Mechanism.* It was operated by a very primitive device, Fig. 86, which was similar to the operating cable and striker already described, except that in this case the striker was attached to the crosshead and the operating cable was attached to a sheave on the valve stem and was carried around an idler sheave at that part of the track farthest from the cylinder. Stop buttons were attached to this cable for the striker to impinge on and thus close the valve, which, upon the return of the piston, was opened by a weight attached by a short chain to another sheave on the valve stem, and so arranged that whichever direction the valve stem was rotated caused the chain to be wound up and the weight lifted, the latter returning

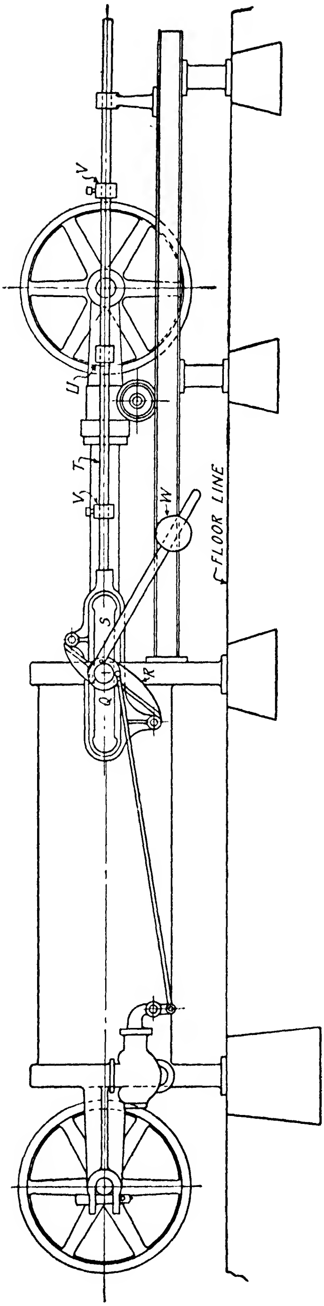



Fig. 87. Hydraulic Push Machine Fitted with Crane Limit Valve

to its original position as soon as the power which lifted it was relaxed.

**Crane Limit Valve.** Another limit valve, that of the Crane Company of Chicago, will make the list large enough to give a good general idea of the principles and construction of this feature of the hydraulic elevator. The operating mechanism, Fig. 87, resembles the Whittier in that it does not completely shut off the ingress or egress of water to or from the cylinder, but merely chokes it, and the initial start-up in either direction is effected by the water which leaks through the limit valve, when the operating valve is opened. The actual stoppage of the piston is brought about by its coming in contact with a barrier placed to resist its further travel, so that the entire strain is received by the cylinder, reinforced, however, in this case by bolts and crossbars.

Fig. 88 gives a clear idea of the Crane horizontal machine. It is a sectional plan of the cylinder, piston, and crosshead of the Crane push type of horizontal machine, with its reinforcing bars across each end. Those at the closed end are cast as part of the cylinder head while that at the open end is a separate piece, perforated in the center for the passage of the piston rod and held in place by two strong bolts on each side of the cylinder and extending their entire length. The piston and crosshead

each have a recess cast in their hubs to receive a rubber ring *A* which acts as a bumper to lessen the shock of impact. The very nature of this arrangement necessitates the making of the cylinder of the exact length to suit the travel of car and the ratio of gearing, no material change being-possible after the installation is once made.

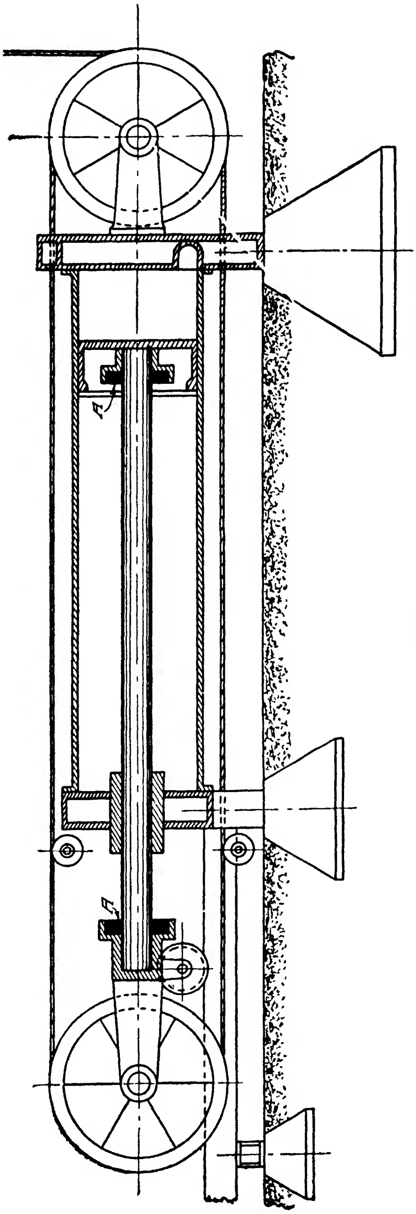



Fig. 88. Details of Cylinder and Piston Arrangement of Crane Machine, Showing Rubber Bumpers

*Operating Mechanism.* Several types of limit valves are shown in Fig. 89. The Crane limit valve is shown in section in Fig. 89*a* and the operating mechanism has already been referred to in Fig. 87. The valve stem is operated by a rock arm, the upper end of which works in a link or yoke in the end of the valve stem. The lower end of the rock arm is operated by a rod connected to another rock arm *R*, which vibrates on a pin attached to the side of the cylinder at the open end. On this pin, sliding horizontally, is a frame or long link *S*, from one end of which extends a rod or shaft *T* running full length of the track on which the crosshead travels. To the crosshead is attached an arm *U*, which resembles that in the Whittier machine, Fig. 89*b*, and with a similar office,

namely, to strike the collars $VV$, which are on the shaft $T$. These are so set that the arm $U$ will strike either one of them as it approaches the limit of its travel at either end of the stroke, the

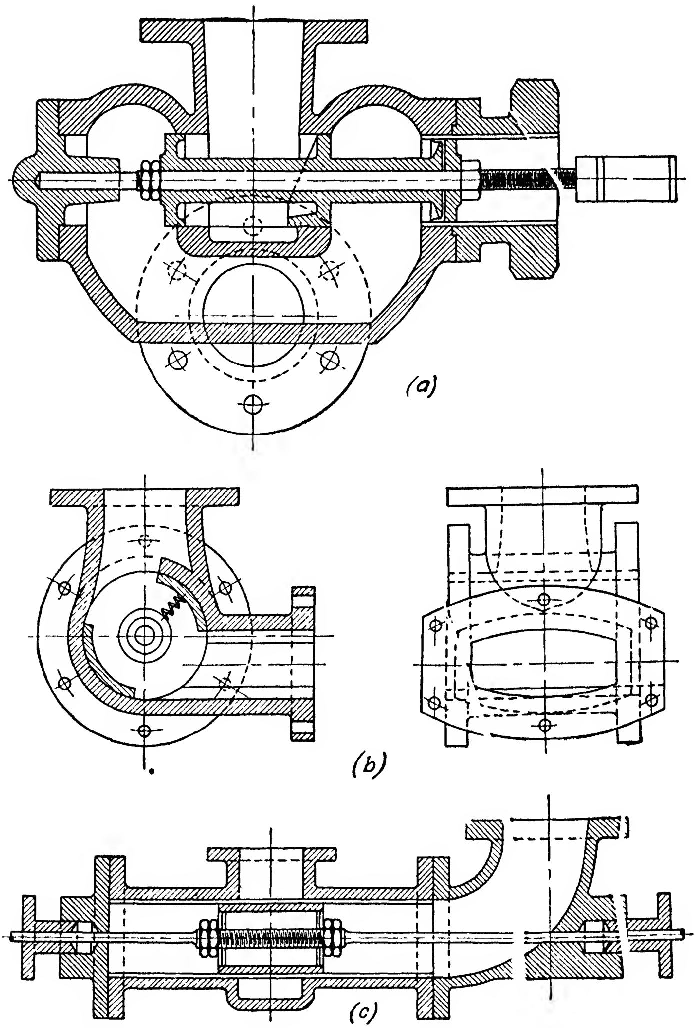



Fig. 89. Types of Limit Valves (*a*) Whittier Type; (*b*) Crane Type; (*c*) Moore and Wyman Type

effect being to carry the rod along with it. The open part of the link slides on the pin on which the vibrating lever or cam $R$ rocks. This link has two brackets, one above and one below, which carry

rollers. As this link travels along horizontally in either direction, one or the other of these rollers engages with the rock arm, the effect being to make it vibrate slightly, the vibration being always the same in whichever direction the rod moves. The action of the rock arm closes the valve and, upon the return of the piston, the weight *W* on its arm brings the valve back to its neutral position with the valve open.

This valve, like that of the Whittier machine, is not a limit valve in the strict sense of the term, but simply a retarder, diminishing the speed of the piston and lessening the shock against the real limit stop, which is the bar across the open end of the cylinder. This arrangement has the disadvantage of being very slow to start in the opposite direction, as the machine is then started only by the water which leaks through the limit valve, until the piston has traveled far enough to allow the weight *W* to open the limit valve again.

## COMPRESSION TANKS

**Necessity for Use.** There are only a few cities in the United States where the pressure in the water mains reaches 100 pounds per square inch; in fact, 60 pounds is an unusual pressure. Hence a compression tank has to be used where a speed equalling that of the steam elevator—about 250 feet per minute—is desired. Roof tanks were frequently used, but at the time the hydraulic elevator made its appearance very few buildings were 100 feet in height and a tank placed on the roof of one of these buildings would produce a pressure of only 45 to 50 pounds in the basement.

**Construction.** The pressure desired was 150 to 200 pounds per square inch. The compression tank made this available. These tanks were usually made large enough to contain sufficient water to fill the cylinders dependent upon them about five times. The water in these tanks was kept under pressure by maintaining in them a large amount of air under pressure. When the tank was filled to the maximum of its designed water capacity, about one-third of the volume of the tank would be occupied by the air under pressure. The water inlet and feed pipes entered at the bottom of the tank so as to reduce the possibility of air in any considerable amount entering the cylinders and producing a jumping motion of the car in starting and stopping. However, cocks were inserted at the highest

points of the cylinders to permit the letting out of air whenever it accumulated in any considerable quantity, for even without these compression tanks this difficulty was experienced because of the air contained in water from city mains.

These tanks were supplied with water by a duplex steam pump to avoid any dead point, as it was necessary for them to start up instantly at any moment. They were governed by a device called a pump governor, which was simply a disk or diaphragm of rubber reinforced by a strong spiral spring and placed in an enlargement of a pipe which ran to the compression tank. This pipe, at its other end near the governor, connected with the steam supply by means of a valve, the stem of which was attached to the center of the diaphragm. When the water pressure reached a certain height, its pressure on the diaphragm closed the steam-supply valve and the pump would stop, and *vice versa*. When the pressure in the tank fell, the spring would open the valve and start the pump again, this part of the operation being absolutely automatic so long as a suitable pressure of steam was maintained at the boilers.

These tanks were supplied also with a valve and pipe to empty them for cleansing and repairs, while to replenish the water, a surge tank was supplied, into which the water was discharged after it had been used in the cylinder. The suction pipe from the pump was led into this tank, and a small air pump was used for replenishing the air. This pump was also operated by steam but was not automatically controlled, as its services were required only every few days where all the joints and seams were tight. These tanks varied in size according to the size and number of elevators to be supplied. Those for small elevators were frequently set on end, but the larger ones were usually set horizontally in cast-iron saddles, which in turn rested on piers of masonry. Sometimes they were set on steel beams above the roof; but no particular advantage accrued to the latter arrangement except a saving of room and possibly less strain on the tank, for there was always an additional gain in pressure at the cylinder due to the height of the tank above it, but the work for the pump remained the same.

**Operation.** The pressure of the air was, of course, transmitted to the water, thus giving the required pressure. As water was withdrawn from the cylinder, the air would expand with a reduction

in pressure. However, the water-pump governor was usually set to act under a variation of not more than five pounds in pressure. Hence, the air pressure, and consequently the water pressure, remained almost constant, the water pump replenishing the water at the end of each trip.

## PILOT VALVES

**Function of Pilot Valve.** With the increased pressure thus obtained it was found that speeds beyond any that had been hitherto attained were possible; but it was also discovered that it was next to impossible to control the elevators at these high speeds by means of the hand rope then in use. The stops were very uncertain and irregular, especially when air, even to a very limited extent, was present in the cylinder. Efforts made to remedy this difficulty resulted in the production of a type of operating valve which was called the pilot valve.

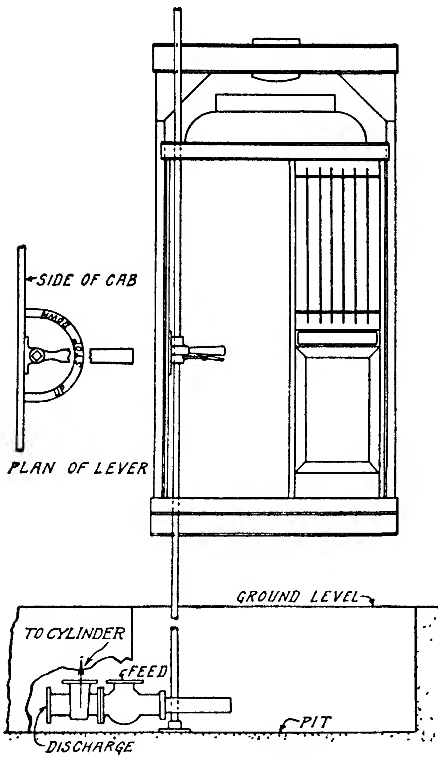


Fig. 90. First Type of Lever Control

**Early Efforts.** The first step in the direction of speed control was very crude and consisted in using a square shaft set vertically and running the entire length of the elevator shaft from the ground to the roof, passing through the elevator cab, Fig. 90. A forked bracket was fastened securely to the inside of the cab and between the forks of this bracket a lever was placed, with a square hole in it which surrounded the square shaft. It was a sort of box wrench and, as the car traveled up and down the shaft, this wrench went with it. By turning this wrench a corresponding movement of the shaft could be produced.

At the lower end of this square shaft a pinion which meshed in the teeth of the valve rack was fastened so that by turning this handle in the cab, the valve plunger could be made to travel lengthwise of the valve barrel and thus turn on or off the water. Attached to the bracket which held this wrench or lever was a quadrant on which were the words: Up—Stop—Down. This was to indicate to the operator which way to move the lever in order to produce the desired effect, for if the handle were moved to cover any one of these words the proper result would be obtained.

This arrangement, while a decided improvement over the hand or operating cable, still fell far short of what was desired and had

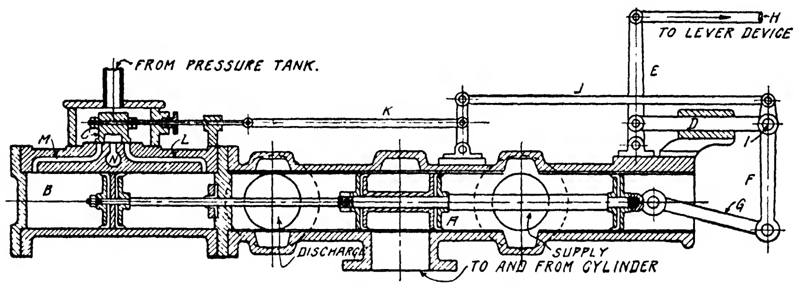



Fig. 91. Early Type of Pilot Valve

many undesirable features, the principal ones being the tendency of the shaft to bow out of the perpendicular in high runs and the friction and rubbing sensation thereby induced in the cab.

**First Pilot Valve.** *Construction.* The efforts of those interested resulted eventually in the production of the valve shown in Fig. 91. An examination of this device shows that it does not differ in the main body from the valves used heretofore, but that certain additions to it made its operation unique as compared with those formerly in use. The principal change was the addition of an auxiliary cylinder to the end of the ordinary valve barrel. This cylinder was fitted with a piston, the piston rod being an extension of the valve stem.

This cylinder was supplied with three ports, a slide valve, and passages for the actuating force, as in the case of the cylinder for a steam engine. The force used, however, was water from the compression tank, the ingenious and novel feature of the apparatus

being the valve motion. Referring to Fig. 91, *A* is the operating valve for admitting water to, and discharging it from, the cylinder which actuates the elevator. *B* is the cylinder of the pilot valve with its piston attached to the extension of the stem of the operating valve. *C* is the small slide valve for admitting water under pressure to either end of this cylinder as desired. *D, E,* and *F* are levers for moving and controlling the movement of the slide valve *C*, while *G* is the connection by which the plunger of the operating valve returns the valve *D* to its normal position as soon as the plunger has reached the position it is made to assume through the movement of *E*. The lever *E* is connected to the operating lever in the cab by the rod *H* and two cables, all of which will be fully described later.

*Operation.* If the operating lever in the cab is in its central or "stop" position, the operating valve will be exactly as shown here in Fig. 91. Should the operating lever be moved either way, the effect will be to move *E* a corresponding amount. This will cause *D* to slide endwise through its bearing, carrying with it the valve rods *J* and *K*, thus sliding the valve *C* on its seat until it opens an end port to the cylinder *B*. Hence water under pressure will be admitted to that end of the cylinder *B*, causing the piston to move away from it. The piston, being attached to the plunger of the operating valve, carries the plunger with it and thus opens the operating valve for either the admission or discharge of water, according to the direction in which it moves.

The stem of the plunger is connected to the lower end of the lever *F* by means of the connecting rod *G*, and as it moves it carries the lower end of *F* with it. This being pivoted at *I* restores the valve *C* to its first position, covering both end ports to cylinder *B* and thus stopping the valve plunger. The operating valve remains open until the lever in the cab is restored to its central position. This in turn moves *E* back to the vertical position, but this movement slides *C* along on its seat in the opposite direction, thus opening the port to the other end of the cylinder *B* and admitting water to the other side of the piston, driving it back toward the position shown in the figure. The slide valve *C*, while it opens the end port to the pressure tank, also opens the opposite end port to the exhaust or discharge port. This allows the water used in the first movement to escape. When the plunger of the operating valve reaches its central or closed posi-

tion, it has, through the medium of $G$ and $F$, restored the slide valve $C$ to the position shown in the illustration, thus stopping the valve movement, everything being again at rest, including the piston in the cylinder of the hydraulic engine which moves the car.

The levers $E$ and $F$ are so proportioned to the stroke of the operating valve that if $E$ is moved only enough to cause $C$ to open the end port $M$ or $L$ but halfway, the main plunger will make only one-half stroke before restoring $C$ to the position where it closes both ports $M$ and $L$, and so in proportion with any fraction of the stroke desired. By this arrangement the operating valve may be opened as much or as little as desired, and by this means any desired speed may be obtained.

The description here given is that of the earliest pilot valve. Many changes, modifications, and improvements have since been introduced, most of which will be fully described later.

## OPERATING-LEVER DEVICE

The mechanism by which the pilot valve was controlled from the cab was fully as ingenious as the valve itself, and was devised at the same time, one being essentially a part of the other.

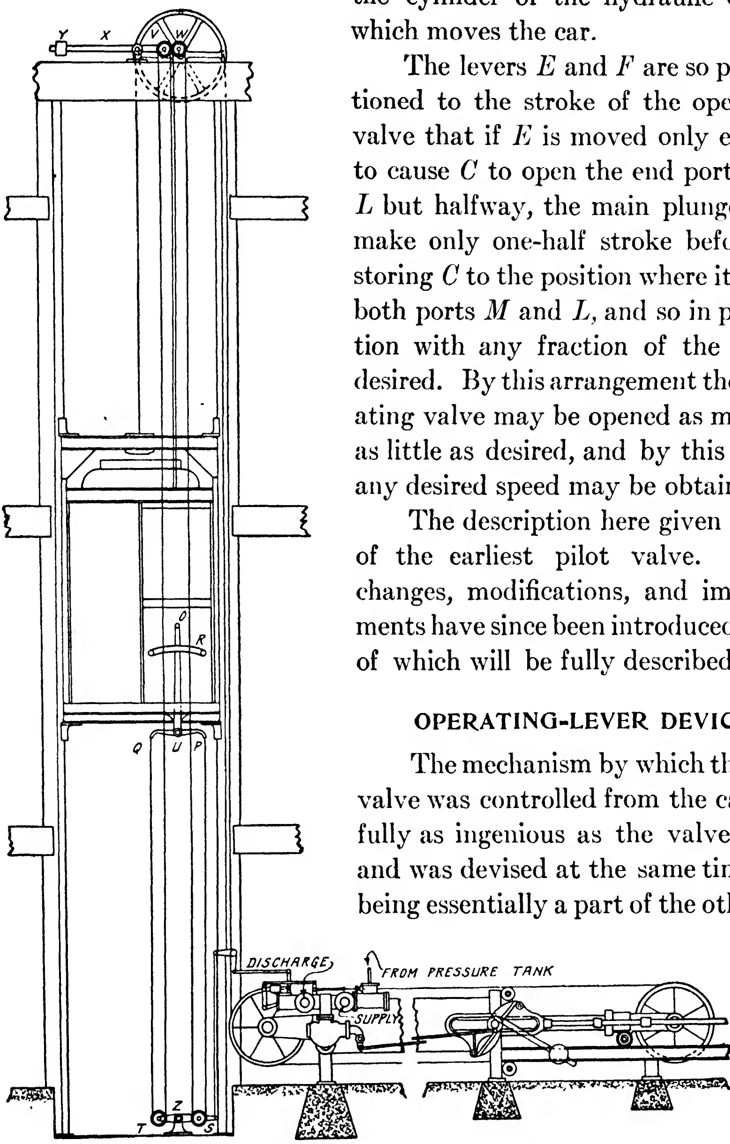



Fig. 92. Crane Operating-Lever Device. First Successful Lever Control for the Elevator Cab

**Crane Type.** The original idea of a lever in the cab, as previously described, was retained, but this time it was set vertically. Referring to Fig. 92, showing the Crane device, *O* is the lever by which the pilot valve is operated. If the lever *O*, which has an inverted **T**-shape, and which is pivoted at *U*, is moved in either direction from the

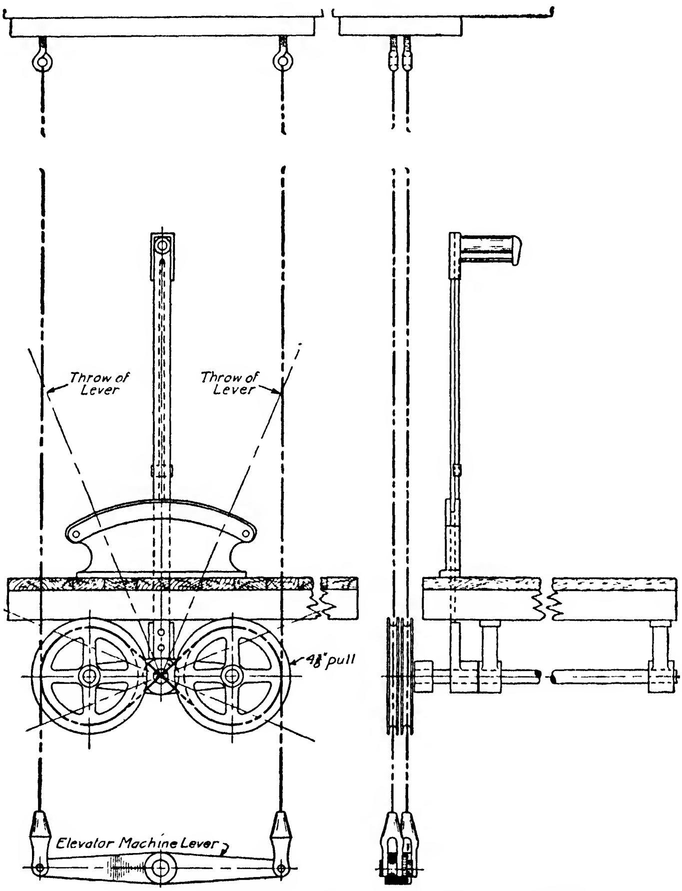



Fig. 93. Operating-Lever Device with Stationary Cables

perpendicular, one extremity of the bottom of the lever, *P* or *Q*, will be correspondingly elevated and the opposite one depressed.

From these points, *P* and *Q*, small cables are led down to and around the sheaves *S* and *T* at the bottom of the hatchway, and thence up and around similar ones, *V* and *W*, at the top of the run,

and then down to the top of the cab, where they are made fast. These sheaves, *V* and *W*, are mounted on a pivoted lever *X*, to the outer end of which an iron weight *Y* is attached for the purpose of keeping the cables taut.

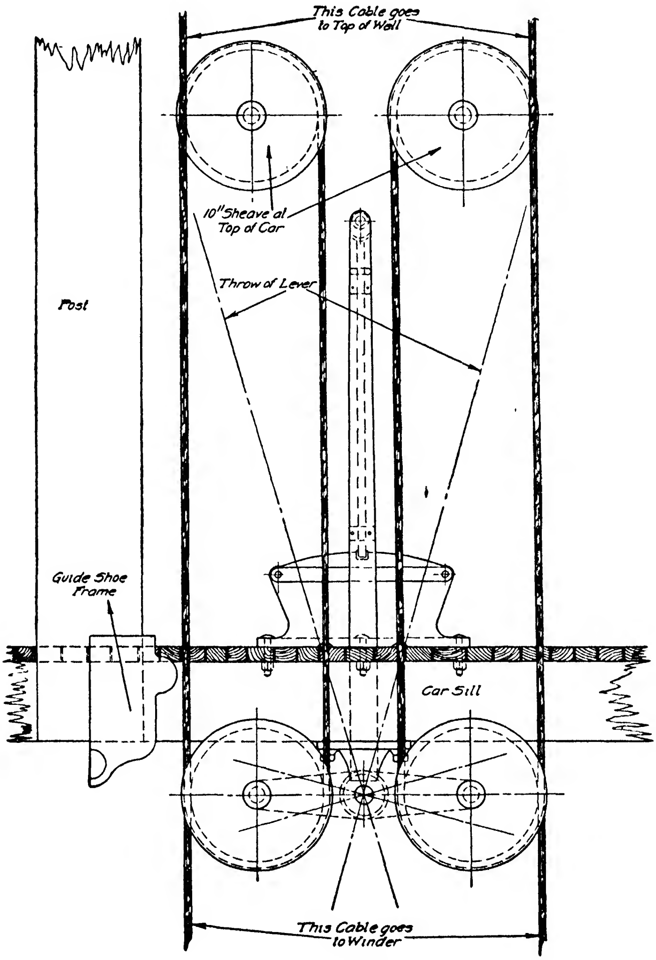



Fig. 94. Operating-Lever Device with Extra Sheaves

When the lever *O* is moved to one side in either direction, and the points *P* and *Q* are respectively raised and lowered, one will take up, and the other give out, a portion of the cable, and the effect will be that the sheaves *S* and *T* at the bottom of the hatchway will be moved

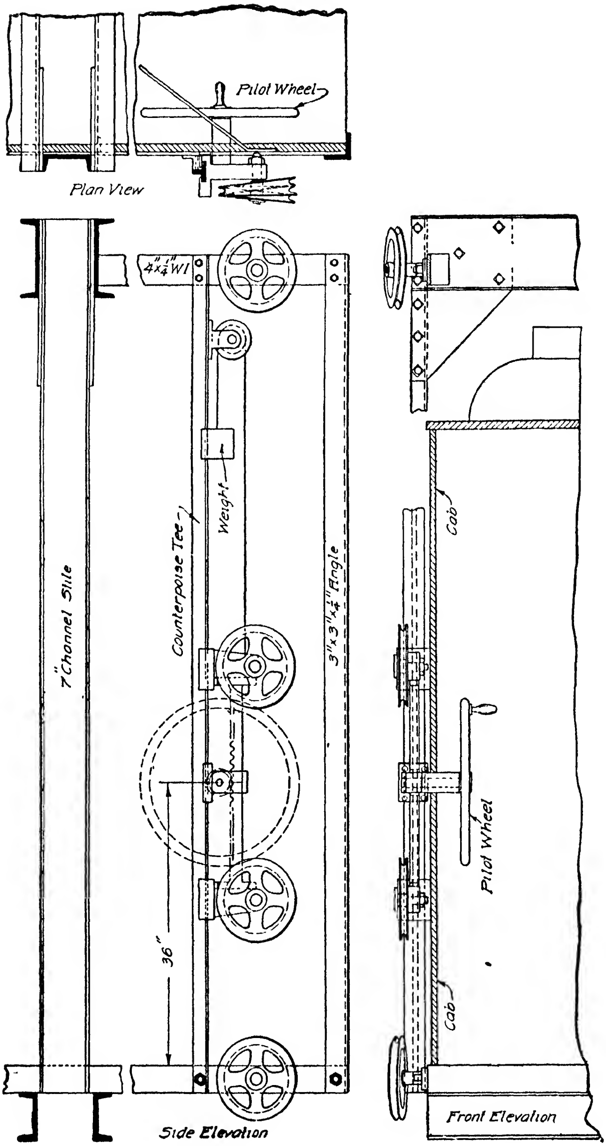



Fig. 95. Pilot-Wheel Operating Device with Rack and Pinion
*Courtesy of Kaestner and Hecht Company, Chicago*

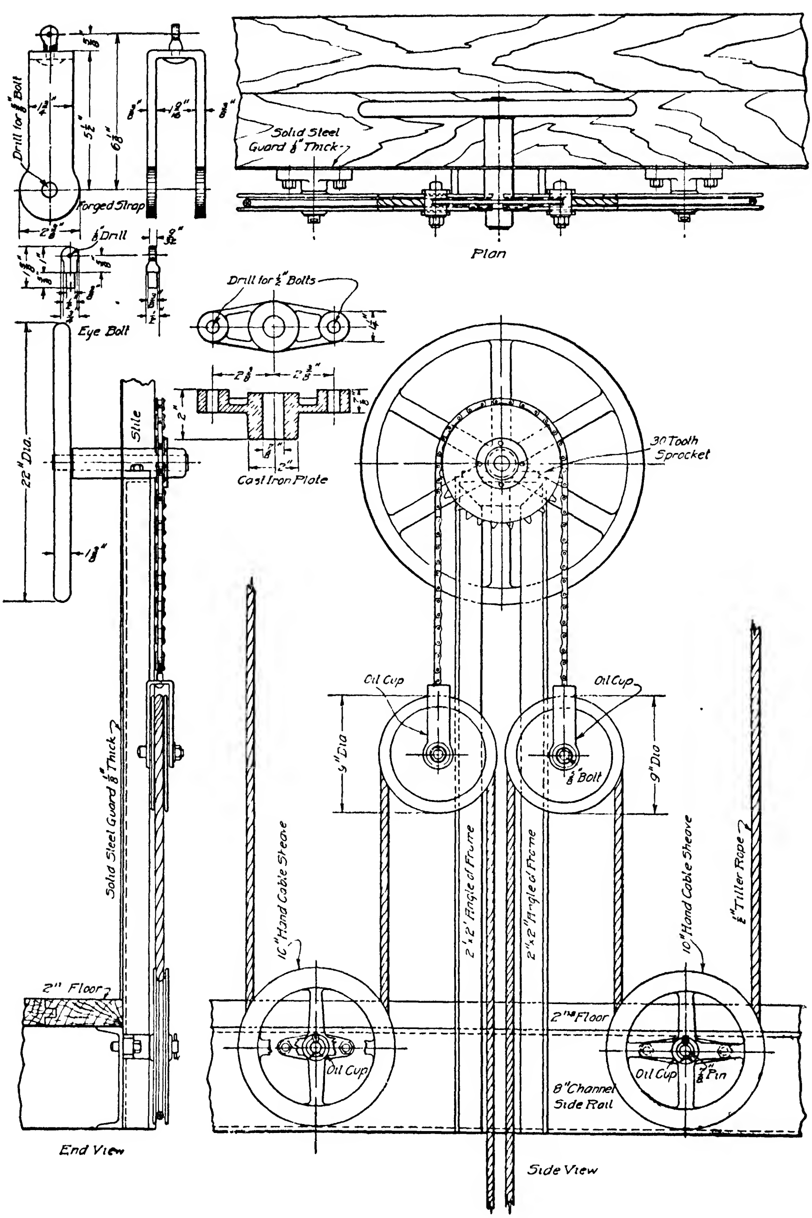



Fig. 96. Pilot-Wheel Operating Device with Chain and Sprocket
*Courtesy of Kaestner and Hecht Company, Chicago*

one up and the other down. Being mounted on the pivoted lever *Z*, they will carry it with them. This lever will move the lever *E* on the pilot, producing the results previously described.

**Other Types.** The Crane arrangement was the first practical lever device for the control of elevators. Several others were afterward developed, most, if not all of them, being arranged to operate with fixed or stationary cables in place of cables moving with the car, Fig. 93. One end of each of the cables was attached to the lever at the bottom of the hatchway or led over sheaves directly to the valve apparatus. The other ends were fastened to a suitable beam at the top of hatchway in the penthouse, long eyebolts being used which were threaded their entire length of two feet or more and supplied with nuts for taking up the slack and keeping the cables taut. Figs. 94 to 96 show several other forms most popular at the time. Figs. 95 and 96 show a form of operating device which came into use at this time which is called the pilot wheel. This had the advantage of a greater range of stroke than the lever, so that it was applicable to the ordinary valve with the operating sheave, rack, and pinion. Simply revolving the wheel one-half turn in either direction would cause the elevator to ascend or descend according to the direction desired, while the neutral or stop position was determined by moving the wheel until a certain mark upon its rim was uppermost. This apparatus required more manual effort than the lever and was not, on this account, suitable for high speeds; in certain localities it became, and still is, very popular.

**High Speeds with Operating-Lever Device.** This method of using the lever in connection with the pilot valve enabled easy starting and gentle stopping to be accomplished at high speeds, and it was only a short time until speeds as high as 600 feet per minute were easily obtained. To reach this speed it was frequently necessary to gear the horizontal machines as high as 10 to 1, the vertical cylinders standing on end in a narrow space alongside the hatchway. There was practically no limit to their length, for the same space would be required for the sheaves, crosshead, and cables. With horizontal cylinders, however, valuable space in the basement of the building was required except where the cylinders were mounted on top of each other. The pressure carried at this time in the compression tanks seldom exceeded 200 pounds to the square inch, and while it was known and admitted that

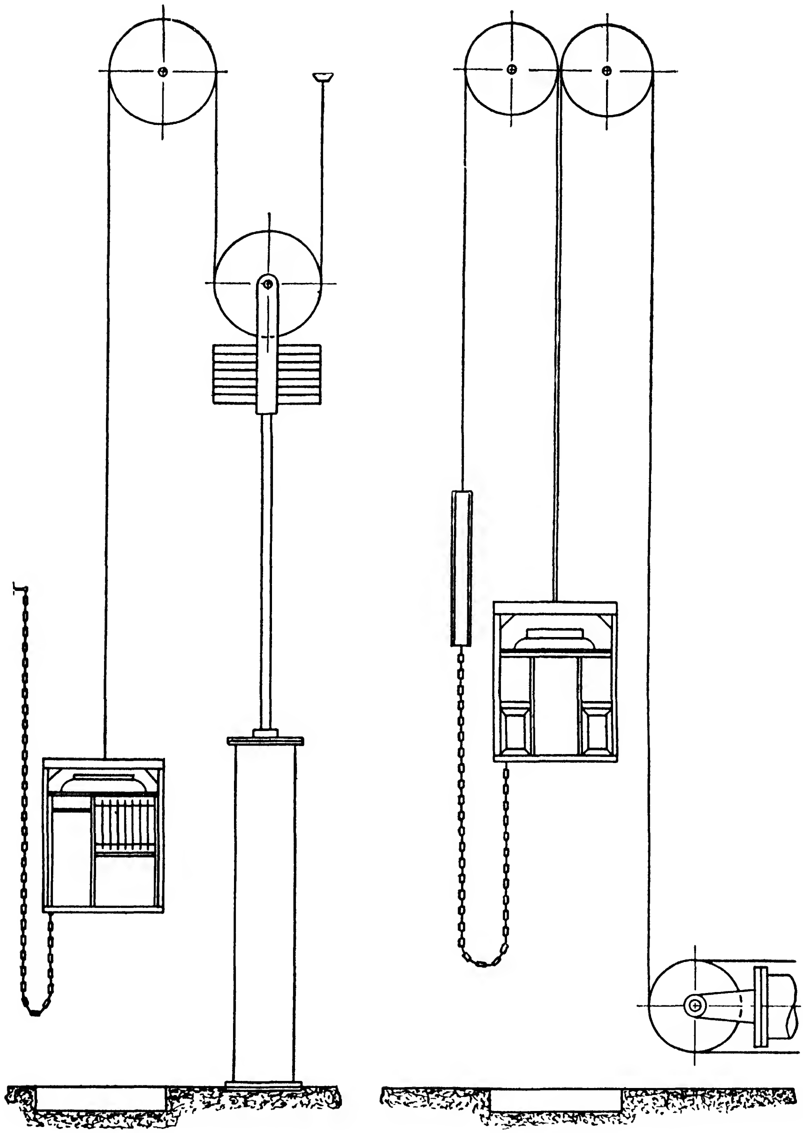


Fig. 97. Diagram Showing Method of Using Counterbalancing Chains

there was a great loss in power caused by friction, due to the use of comparatively large cylinders, no change was made for a long period. The smoothness of operation made the hydraulic elevator seem the acme of perfection. It supplanted the steam elevator in most cases where new plants were contemplated and in many already established.

**Counterbalancing Chains.** In buildings many stories in height, the length and number of cables used became an important factor in the counterpoising of the cars. When about six $\frac{5}{8}$-inch cables were used on a horizontal hydraulic elevator in a building of 15 or more stories, it was found that if the car was counterpoised even to a slight degree, the weight of the car when at the top floor would not be able to overhaul six cables and move the piston as well, the cables alone weighing something over 800 pounds. Compensating chains were then employed, Fig. 97. A straight-link chain was used which was six or seven feet longer than half the travel of the elevator, and of such a size that it would closely approximate the weight of the overbalancing cables. One end of the chain was attached to a point halfway up the hatchway, the other end being fastened to the bottom of the car platform. When the car was at the bottom landing, the entire weight of the chain was supported from the wall of the shaft. When the car was at the top landing, the entire weight of the chain would hang from the car, thus compensating for the weight of the cables leading down from the overhead sheaves to the machine below. Similarly, for any position of the car between the limits of its run, a proportionate amount of the chain would be hanging from the car, the remainder being carried by the fastening in the wall of the shaft.

In case the amount of weight required for compensation was so great as to make the necessary chain too large, two chains, each of half the weight, could be used. To prevent any noise from the rattling of the links, a piece of soft rope was woven in and out through the links for the entire length of the chain.

## HIGH-PRESSURE HYDRAULIC TYPES

**Inefficiency of Former Types.** Although with a water pressure of from 120 to 200 pounds per square inch, very good speeds and capacities were easily attained, the apparatus required was very ungainly and occupied much valuable space. It was undeniable that

a large percentage of the input of power was lost in friction, the principal source of the loss being in the size of the cylinder required and in the cables and high gearing in use.

Efforts had been made to overcome the friction due to the use of many sheaves and cables by using one or more piston rods in a thrust machine. These were of cast iron in the form of toothed racks which geared into pinions on a drum shaft. The element of danger which existed in the risk of a possible breaking of gear teeth made this method of the application of power seem too risky, especially for high runs or for the carrying of passengers, and, although many very successful machines were made and put into operation, they never became very popular.

Elevator builders had for some time felt that if water at a higher pressure was available a great improvement could be effected; but it was not until the introduction of electricity as a motive power for elevators that any effective effort was made in this direction.

**Introduction of High Pressures.** A manufacturer of elevators in Chicago was determined to investigate the use of high pressures in London, England, and for that purpose sent a commission to London to secure all the information possible on the subject. In London, the high pressure was obtained from high-pressure street mains built and controlled by a corporation, but this scheme had already been unsuccessfully promoted in America. The result of the commissioners' visit was the determination to use water at a pressure of 750 to 800 pounds per square inch, and that its production should be made on the premises in each case. While this arrangement resulted in the saving of considerable space when compared with the previous large horizontal machines, there was still considerable room required for the boilers, pumps, and tanks.

The type of hydraulic engine determined on was the vertical, to be located either alongside or to the rear of the hatchway. As the working pressure was very much greater than that of the low-pressure system, the consequent reduction in the diameter of cylinder required was great enough to permit the pumping engine being located near-by.

*Difficulties Experienced.* Many difficulties were encountered. Fittings on the feed pipes were hard to keep tight; air leaked out of the pressure tanks; and air at the pressure required was hard to pump. In addition, it was found that the pressure tanks required for the

supply of such small cylinders were so proportionally small that there was great fluctuation in the pressure. Accumulators were therefore used. Then it was found that the old cup packings for piston and valves were inadequate. In the case of the pistons, plaited square hemp kept in position by a brass follower ring was introduced as a remedy. With the valves it was found that the passage of water at the high speed, consequent upon the reduced size of openings, wore the holes in the brass linings of the valves so rapidly that it was difficult to maintain a proper graduation. The brass perforated linings were abandoned, and ports or full openings used. This necessitated the use of valve plungers of a different construction.

Still another difficulty was experienced, namely, the operation of the pilot valve under the increased pressure. For while the valve casings, the cylinder, and the pipes and fittings had to be materially heavier in order to withstand the increase in the water pressure used, this same cause necessitated a reduced area in the openings feeding both valves and cylinders. In the case of the pilot valves, as a result of this reduction these openings often became clogged or, in fact, entirely closed by small pieces of waste or canvas from the pump piston packings. This occurred so frequently as to become a constant source of annoyance and also to detract seriously from the advantages of the high-pressure system.

**Improved Pilot Valve.** The difficulty was overcome at last by a Mr. H. F. Witte of the Otis Elevator Company, Chicago. He suggested the use of a separate tank for the operation of the pilot valve, this tank to carry a lower pressure, say 80 to 100 pounds. Upon being tried, it proved to be all that could be desired. It was, in fact, simply a return to the old conditions of low pressure so far as the pilot was concerned. Fig. 98 gives a sectional view of one type of this valve which is in general use.

*Construction.* It will be observed that the proportions differ noticeably from those of the low-pressure valve, the motor being much larger in comparison with the operating valve. This, it must be remembered, is due to the fact that the pilot and motor are to work on low pressure, while the operating valve is designed for the higher pressure.

A further inspection of Fig. 98 will show that, while the levers for operating the pilot are similar to those used for the low-pressure

pilot, the valve, in this case, is differently constructed, in that it is a sort of piston or plunger. Instead of two pipes being connected with the casing of the pilot, there are three. The middle pipe *A* is the

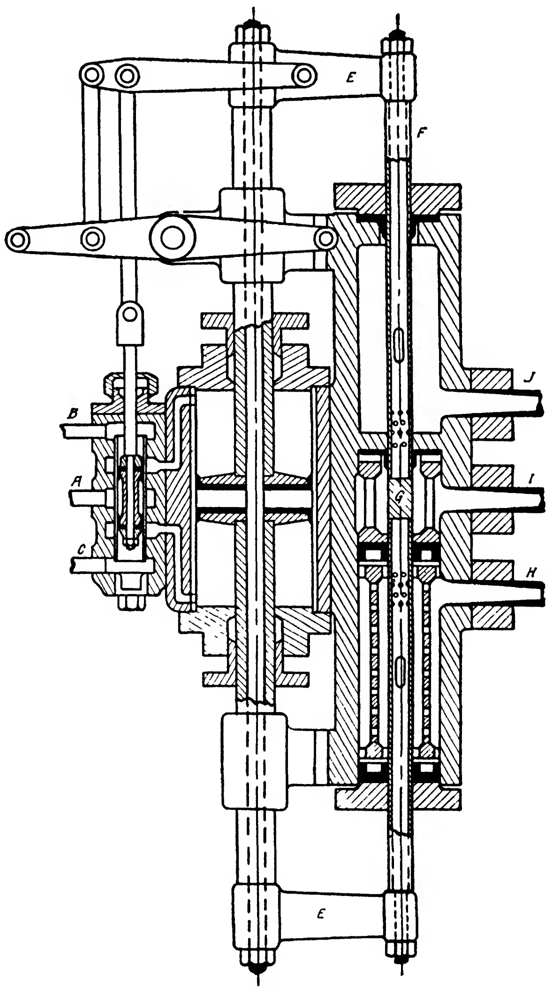



Fig. 98. Section of Witte Improved Pilot Valve

feed, the two outer pipes *B* and *C* being for the discharge from either end of the motor cylinder.

*Operation.* The motion is exactly the same as in the low pressure. For example, the pilot valve admits water through *A* to the

lower end of the pilot cylinder and at the same time allows a discharge from the upper end through $B$ and *vice versa.* The levers and their connections are the same as in the other valve, but the pilot is, in this case, attached to one side of the operating valve instead of at its end. The connections with the operating valve are also different, for here both ends of the piston rod of the pilot are connected with

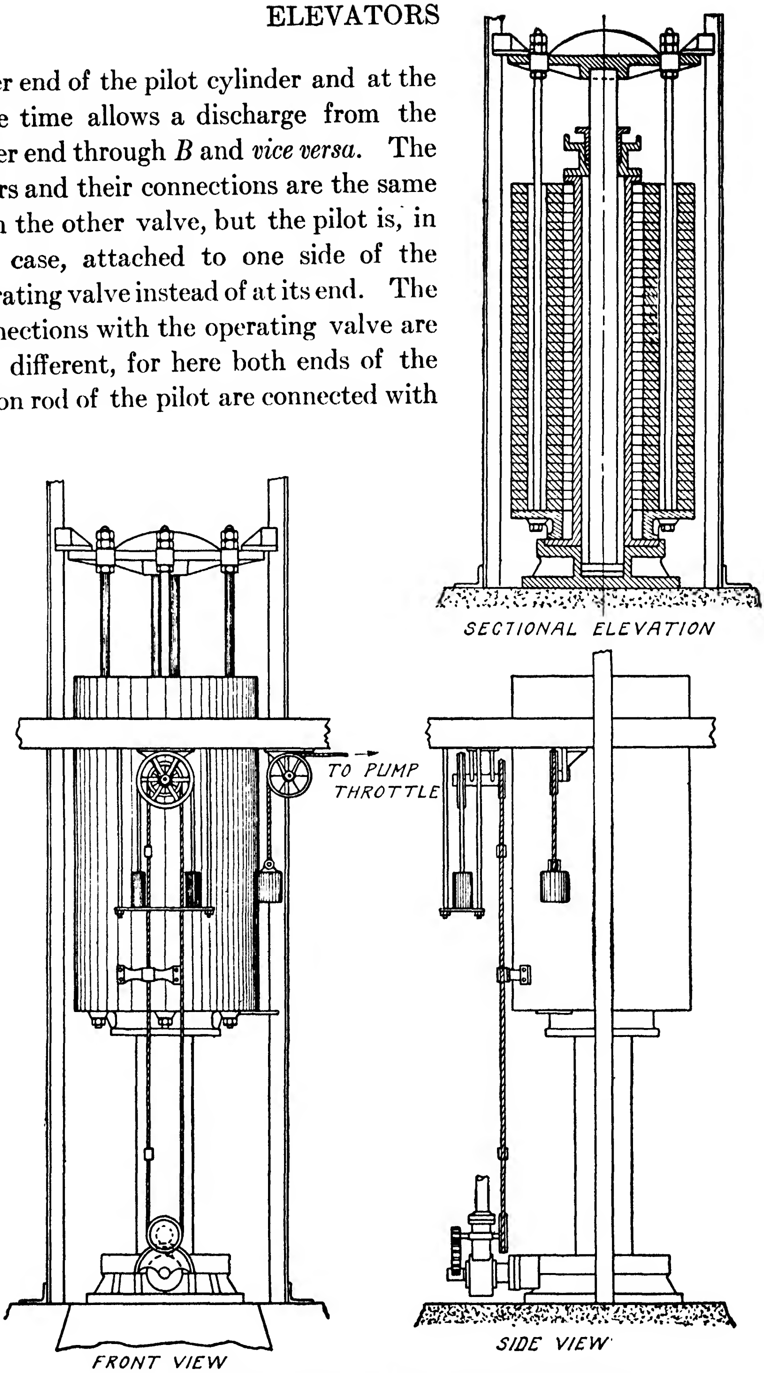



**Fig. 99. Details of Accumulator Used to Replace Pressure Tanks**

the operating valve by means of the arms $EE$. $F$ is really the operating valve itself, being a tube, solid in the center at $G$, with perforations in it for the admission of water to its interior. The port $H$, in the valve casing, is the feed from the accumulator. Port $I$ leads to and from the cylinder, while $J$ is the discharge.

If, through the action of the pilot, the valve $F$ passes upward through the valve casing, the perforations in its lower half are brought into position so as to allow water from the accumulator, Fig. 99, to pass into the port $I$ and thence to the cylinder. Should the motion of the pilot be reversed, the operating valve $F$ will, when sufficiently depressed, permit the water in the cylinder to escape to the discharge through the port $J$.

A high-pressure installation with a Witte pilot valve, accumulator, and other auxiliary apparatus is given in Fig. 100.

**Use of Accumulator.** With the introduction of the accumulator, a different method of feeding the operating cylinders of the hydraulic engines was adopted. In using the tanks it had been the custom to make them large enough to hold sufficient water for two or more trips without a very great drop in the pressure, but it was not found feasible to do this with the accumulator, especially where several elevators were in operation on the same system. So the accumulators were made of sufficient capacity for one elevator to run, say, one trip, and for two or more elevators, the increase in capacity was not proportional to the increase in the number of elevators. The idea was to keep the pumps working continuously or nearly so. By so doing, the elevator was really operated by pumping the water directly into the cylinder. Between times, the water discharged by the pump went into the accumulator, which thus acted as an equalizer by supplying water when the pump was slow in delivery, and receiving and storing it up for future use when the elevator was shut down. This arrangement was particularly suitable where a number of elevators were operated on the same system, for the pump was so proportioned that when running at a constant speed it would be just sufficient to supply all the elevators, any differences in demand and supply being taken care of by the accumulator.

**Changes Made in Pumps.** The operation of the duplex pump, such as was used to supply the low-pressure tank, was found to be

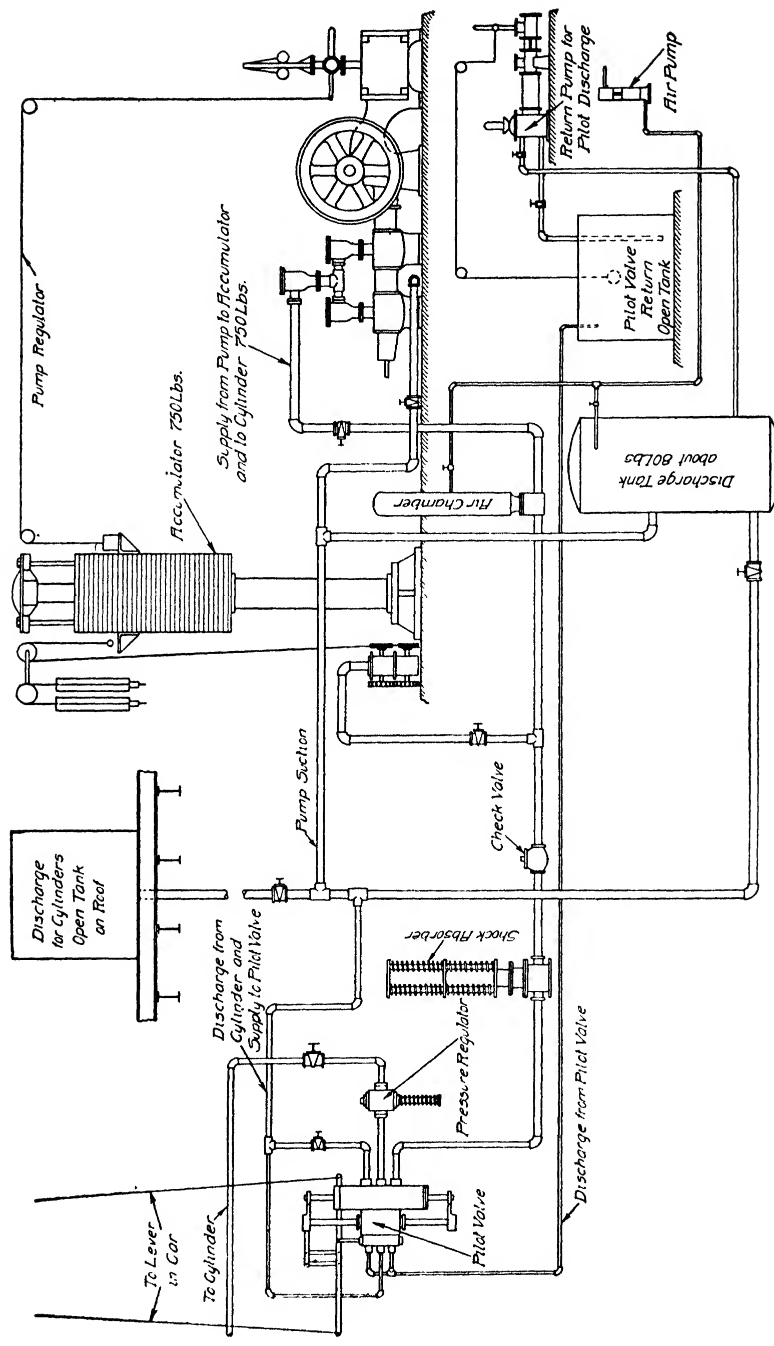



Fig. 100. Diagram of Typical High-Pressure Hydraulic Installation with Accumulator, Roof Tank for Pilot Valve, Shock Absorber, Etc.

unsuitable for this type of equipment because the pulsations of the water pistons could be distinctly felt at the platform, although the pumps were provided with air chambers. The reason for this was that the stroke of the pump was not even throughout, being accelerated at one portion and retarded at another. To remedy this pumps were built to operate with flywheels, thus relieving the apparatus of most of the disagreeable pulsations. (See Fig. 100.)

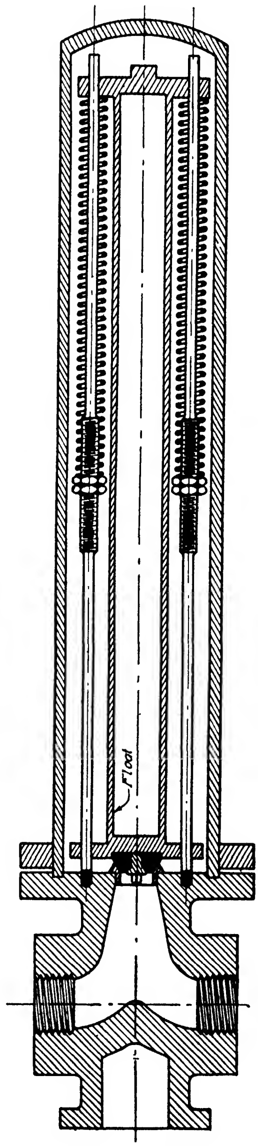


Fig. 101. Air Chamber to Absorb Pulsations of Pump

**Introduction of Air Chamber.** A special air chamber, Fig. 101, was devised to eliminate what remained of the pulsating effect. It is simply an air chamber made in the usual manner but suitable for high pressure, and having within it a tube closed at both ends and held down on a seat at its lower end by the air pressure in the chamber. Two spiral springs are used to neutralize the weight of the float, thus making the valve more sensitive. These springs can be adjusted to give more or less pressure to the tube, which, when seated, prevents egress of either air or water from the chamber. When properly adjusted, this apparatus will operate to reduce any pulsations in the feed pipe, the inner tube rising and falling with any unevenness of pressure, even though it be but momentary. When at rest on its seat, the inner tube prevents the escape of air into the feed pipe, and thence into the cylinder, where its presence produces an unpleasant jumping of the car.

**Shock Absorbers.** Water is practically incompressible, that is to say, it has not the elasticity possessed by air, hence if a volume of water traveling through a pipe at a high velocity is suddenly stopped, its momentum and weight, together with its inelasticity, combine to cause it to produce what is called a

"water hammer" in the pipes. This is likely to cause a fracture in some of the fittings, and it also tends to displace the packing and damage the valves, being an undesirable feature in either high- or low-pressure systems. In the low-pressure piping it is very well taken care of by the air chamber, but in the high-pressure piping a special arrangement is employed.

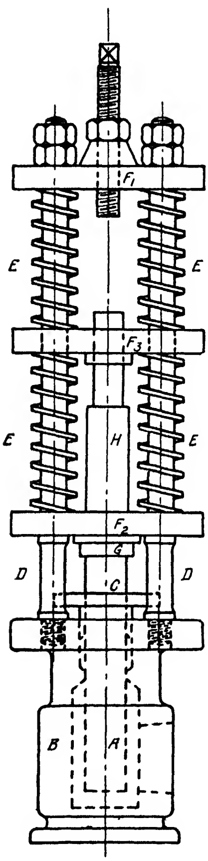


Fig. 102. Details of Shock Absorber

*Construction.* It will be seen by referring to Fig. 102, that the shock absorber comprises a short plunger or ram $A$ of comparatively small area fitted with a casing $B$ and a stuffing box $C$. At each side of the stuffing box are pillars or strong studs $D$, two sets of spiral springs $E$ and three plates $F_1$ $F_2$ and $F_3$, which are so arranged on the studs as to divide the springs, the upper ones being much heavier than the lower ones. The ram $A$ is also made with two shoulders located at $G$ and $H$ for engaging the plates $F_2$ and $F_3$.

*Operation.* When the column of water traveling at high speed is quickly stopped by the shutting of the operating valve, the momentum of the water is expended on the lower end of the ram $A$ and raises it. But the lower shoulder $G$ lifts the lower plate $F_2$, which in turn raises and compresses the lower set of spiral springs. Should they be inadequate to bring the ram to rest, it continues to travel upward until the second shoulder $H$ comes in contact with the under side of the second plate $F_3$, which in turn lifts and compresses the upper and stiffer spiral springs, finally overcoming the momentum of the column of water. When this has been accomplished, the springs return the ram to its first position ready for another and similar operation. The introduction of the shock absorber into a high-pressure system is shown in Fig. 100.

**Machine Arrangement.** *Cylinder and Plunger.* The piston and comparatively light piston rods in vogue with the low-pressure machine were dispensed with and replaced by a ram or plunger working in a vertical cylinder not bored or finished on the inside. The

water was kept from leaking out by means of a stuffing box and of a gland which was bored to fit the outside of the plunger, and which was fitted with hemp packing. The opposite end of the plunger was attached to the crosshead in which the traveling sheaves were mounted. This crosshead or frame was made of steel plates or of channels and fitted with guide shoes running on steel guides attached to the end of the cylinder.

*Sheaves.* Where a number of sheaves were used for the purpose of getting a high multiple, they were not placed side by side on one shaft, but on separate shafts, one ahead of the other, that is, in tandem. This was done to keep the lateral space occupied by the machine as small as possible. The fixed sheaves were set at the opposite end of the cylinder in a similar manner, and the cylinder was provided with lugs or feet by which it was bolted to the wall of the building or to a frame of structural steel. This in turn was attached to the floor beams or to trimmers of the hatchway.

*Counterbalancing Effect.* When the water was admitted to the cylinder, the plunger was forced out downward, hence the weight of the plunger, crosshead, and traveling sheaves always acted with the power that lifted the car, and in cases where the latter was small and the load lifted comparatively light, the plunger, sheaves, and crosshead would, in themselves, be sufficient; but in cases where they were not, provision was made for the addition of counterpoise weights attached to the crosshead to make up the deficiency. It must be remembered that only a fractional part of the entire weight of these parts helped to counterpoise the car, for the counterbalancing effect was equal to the total weight divided by the multiple, or number of sheaves on the machine. Thus, if the engine had 6 sheaves, the effective counterpoise of these parts of the engine was only one-sixth of their weight.

## REVERSE-PLUNGER TYPES

One of the machines which resulted from the efforts toward improvement it is well to describe, because it really has merit; its lack of popularity in the early form was due to a failure of one of the appliances connected with it rather than to any inherent defect in the machine itself. The inception of this machine, which is known as the reverse-plunger type, was the cause of an important change

in the arrangement of the low-pressure tank for the operation of the pilot valve.

**Early Type.** *Construction.* It will be remembered that the introduction of the low-pressure tank was found necessary for the successful operation of the pilot valve in connection with the high-pressure elevator. This machine had for its object, or rather for one of its objects, the supply of water to this low-pressure tank, thus dispensing with the pump for that purpose. It comprised the usual plunger machine previously described; but its position was reversed and it was made longer, so that its action was really two to one, the stroke of the plunger being half the travel of the car. The plunger emerged from the top end of the cylinder, the fixed ends of the cables being attached near the roof of the building. The sheave in the cross-head ran in the bight or loop of the cables, which passed under and around this sheave and thence up to a sheave at the top of the hatch, and over this down to the car.

*Operation.* The peculiarity of this machine lay in the fact that the plunger had to enter the cylinder in order to lift the car. Of course, in doing this it was discharging water; therefore it had to lift the car solely by its own weight, the lowering being done by admitting water under pressure to lift the plunger. The plunger was made to discharge the water into the low-pressure tank, and from this tank the high-pressure pump took its supply of water for the high-pressure tank. This feature was a source of economy in power, the added pressure of the supply assisting the high-pressure pump in performing its duties and at the same time dispensing with the low-pressure pump altogether.

*Defect.* The machine worked admirably, and would have become popular but for an unfortunate accident caused by the giving-out of the air in the low-pressure tank. This caused the plunger to descend quickly, thus rushing the car to the top of the shaft with great velocity. This accident, while it condemned the comparatively new machine, led to the adoption of another arrangement which had some good features.

**Improved Type.** The improved type dispensed with the low-pressure air tank and substituted in its place an open tank placed high enough to give the desired pressure for operating the pilot valve without the use of air as an auxiliary. This method is now in general use.

## DEVELOPMENT OF AUXILIARY APPARATUS

**Increased Water Economy.** Of course, during this period of experimenting with the high-pressure hydraulic elevator, the minds of those busy with the improvement of the machine had never lost sight of the idea of economizing in the use of water as a motive power.

*Use of High- and Low-Pressure Tanks.* It occurred to someone that with two tanks, each having a different pressure, some way might be devised whereby either might be used for lifting the load, Fig. 103. To this end a special form of controlling or operating

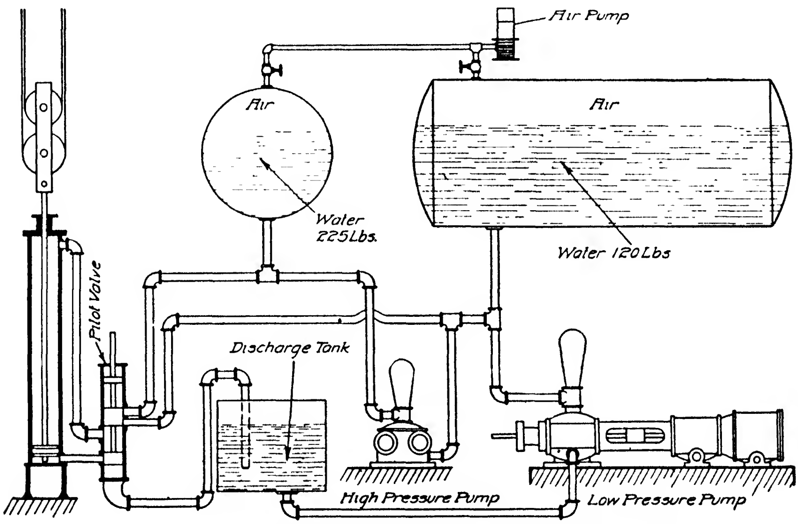



Fig 103. Typical Installation Using High- and Low-Pressure Tanks

valve was devised by which water from either the high- or the low-pressure tank could be admitted to the cylinder as desired, and it proved quite successful. In this connection a feature heretofore unseen developed, for it was found that when using the water from the high-pressure cylinder even with the heaviest loads a certain amount of the water from the low-pressure cylinder flowed in with it, and this was a source of economy quite unexpected. It will be noticed in the diagram, Fig. 103, that no flywheel is provided for either pump, as shown in Fig. 100, the pulsations of the pumps being absorbed by the elasticity of the air above the water in the tanks.

**Improvements in Horizontal Machines.** As high-pressure machines were made horizontal, special features had to be introduced.

*Ram.* To counteract the tendency of the ram to bend inside the cylinder the inner end is fitted with a shoe which rides on the lower side of the cylinder and thus keeps the ram always level. The outer end of the ram is fitted with a crosshead, which carries the traveling sheaves, and which in turn slides upon guides set exactly in line with the cylinder in a manner similar to the guides of a steam engine. The ram does not fit the cylinder for its whole length, but runs through a stuffing box and acts as a piston. Beyond the stuffing box, buffers or bumpers project on each side. These bumpers are exactly in line with bosses on the crosshead, which come in contact with the buffers at the end of the down stroke and prevent the end of the ram from breaking the cylinder head.

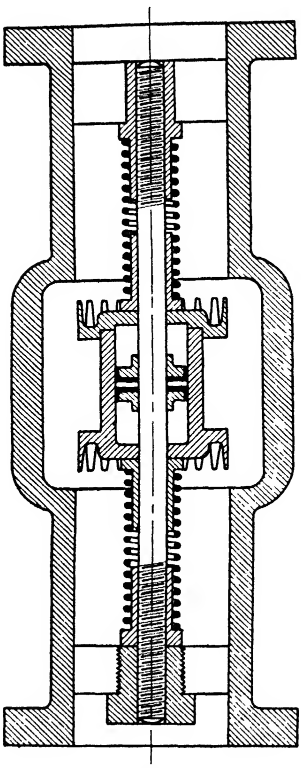


Fig. 104. Section of Speed Regulator

*Stop Mechanism.* The limit stop valves on this machine are very similar to those used on the horizontal low-pressure engine, except that they are of heavier design and of less area. The stopping mechanism is also similar, being the usual wire rope running over two sheaves, one sheave attached to the valve spindle and the other being an idler simply to carry the rope. Instead of being placed at the extreme end of the machine, the idler pulley is set slightly beyond the middle point of the run or travel of the crosshead. Below the lower part of this rope a long rod is placed, sliding in brackets attached to the frame of the machine. These brackets serve as guides as well as supports for the rod. The end of the rod nearest to the limit valve is fitted with an arm which is made fast to the lower part of the horizontal cable which operates the limit valve. The other part of the rod has the two buttons or stops. The arm which is attached to the crosshead and forms the striker slides on this rod and strikes the buttons at the limit of the stroke at either end. The reason for using this

shorter cable is to guard against its coming off the sheave. The operating valve is similar in design to that used with the vertical machine.

**Introduction of Speed Regulator.** With both these high-pressure, high-speed machines, a type of valve is used which is not found to be necessary with the low-pressure machine, namely, the speed regulator, Fig. 104. It consists of a valve set on a spindle in the feed pipe and adjusted to a central position by means of a spiral spring at either end. That portion of the pipe where the valve is located is enlarged to permit of the free passage of the water around it, depending for its action entirely on the velocity of the water passing through the pipe. Should the speed of flow of the water through the pipe exceed that for which the springs are adjusted, the water will carry the valve along with it, thereby partly obstructing the flow of water, and consequently reducing its velocity. As soon as this occurs the springs gradually restore the valve to its normal position, thus allowing more space for the passage of water. The speed regulator is used to prevent a too sudden starting of the car, and also to restrain its speed, thereby keeping it normal regardless of the improper manipulation of the operating valve by the man in the car.

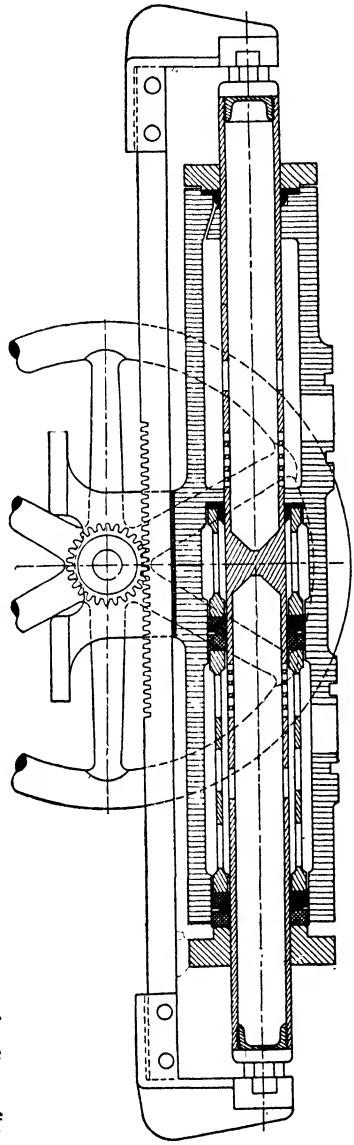


Fig. 105. Section Showing Pilot-Wheel Operation of Control Valve for High Pressure

*Courtesy of Kaestner and Hecht Company, Chicago*

**Development of New Methods of Control.** *Pilot-Wheel Operation.* The controlling valve mentioned as being used with the high-pressure hydraulic

has been spoken of thus far as being operated by the mechanism of the lever and pilot attachment. In many parts of the country where the pilot wheel is the favorite method of control, a valve has been used which closely resembles in construction the valve heretofore considered, Fig. 105. The only difference lies in the discarding of the pilot attachment and the substitution of a rack and pinion, together with the use of a cable sheave on the pinion shaft. The operating cable is passed around this sheave and made fast thereto. It then passes over suitable sheaves to and up through the hatchway, where it is moved by the pilot-wheel mechanism.

This operating arrangement is found to be very satisfactory, one of the chief advantages being the elimination of all the troubles incident to the use of the pilot—namely, the stopping of the tiny water passages by lint and pieces of canvas fiber from the piston packing of the pumps—and the discarding of the low-pressure tank for its operation. But the majority of those who use elevators do not like the manipulation of the pilot wheel, as most of the wheels require from one-half to three-quarters of a revolution in either direction to produce the up or down motion as desired. This requires a much greater amount of physical labor on the part of the operator and the stops are more difficult to make. The lever, with its comparatively short stroke of less than one foot in either direction, makes an easier control and requires less exertion on the part of the operator.

*Electromagnetic Control.* This preference for a simple and easy control led some manufacturers to adopt the electric-car switch for this purpose, but although its application has been successfully accomplished the results are not always what could be desired. The pilot valve as heretofore described is used, but, in place of using the cables mechanically operated by the lever, a controlling switch exactly similar to those used with the electrically controlled and operated elevator is placed in the car. An electric cable is led from this switch to a suitable point in the runway, the cable having a sufficiently long loop to permit the free travel of the car throughout the shaft.

From the cable end, permanent wires are run to electromagnets, which are set on the valve proper in such positions that they

operate the lever when energized. The current necessary for the operation of this control is obtained either from a lighting current in the building or from a battery. While this method of operation is a success from the electrical point of view, it is not by any means perfect. The chief objection to its use lies in the fact that with this arrangement only one speed in either direction is attainable. Moreover, there is always a liability to derangement through the wetting of the magnets.

*Magnetic Control with Push-Button Machine.* There is a certain type of elevator with which this form of control has been used extensively, namely, the slow-speed, push-button-controlled hydraulic machine. In some factory buildings, and in localities where electric power is not available for elevator work, but water under pressure is plentiful, elevators of this type have been installed in cases where no attendant was desired and in places where the elevators were used by employes indiscriminately.

The machines have not given entire satisfaction, owing to the fact that it is next to impossible to keep them perfectly tight for any length of time. Should the piston or the packing around the plunger or any of the valve packings leak even slightly, the car will not stand where it is left, but will slowly creep. This will bring about a very awkward condition for, although the car may move only a few inches from the landing, it will derange the electrical connections of the operating device to such an extent that the machine will become inoperative.

For example, it is part of the push-button plan of operation that no door can be opened unless the car is present at that landing; also, that when the car is between two landings, the pushing of the button at any landing will neither start nor stop it. These are safety measures adopted to prevent accidents. When, on account of leakage, one of these elevators travels either up or down from the landing for even a short distance, it cannot be operated from any floor, nor can the door be opened. Hence, in such a case, it is necessary for the operator to go to the basement and there manipulate the valve by hand in order to move the car to the landing. Of course, where the traffic is so continuous that the elevators never stand more than a minute out of use, these conditions do not have time to affect its use.

## DIRECT-PLUNGER HYDRAULIC TYPES

The direct-plunger is probably the oldest type of hydraulic elevator, isolated instances of such machines having been known seventy years ago. As its name implies, the plunger is attached directly beneath the platform, which it raises through the medium of water under pressure acting on its lower end, and which it lowers by gravity.

**General Construction.** The machine employs a cylinder, usually of cast iron, set vertically in the ground, its top end flush with the bottom of the pit, which is usually three feet deep, Fig. 106. Its lower end is closed by means of a cylinder head made up with a water-tight joint. The top section of the cylinder contains the nozzle or inlet pipe and the stuffing box. This top section protrudes above the floor of the pit for convenience of access. The plunger is a hollow tube, now made of steel but formerly of cast iron. Where the height of travel is great the plunger is made in sections connected by means of male and female threads. Some makers make the male and female threads on alternate ends of each section, but the popular way is to make both ends of each section with female threads, a thimble with male threads being used to connect the sections. On the top of the plunger the platform is placed and fastened.

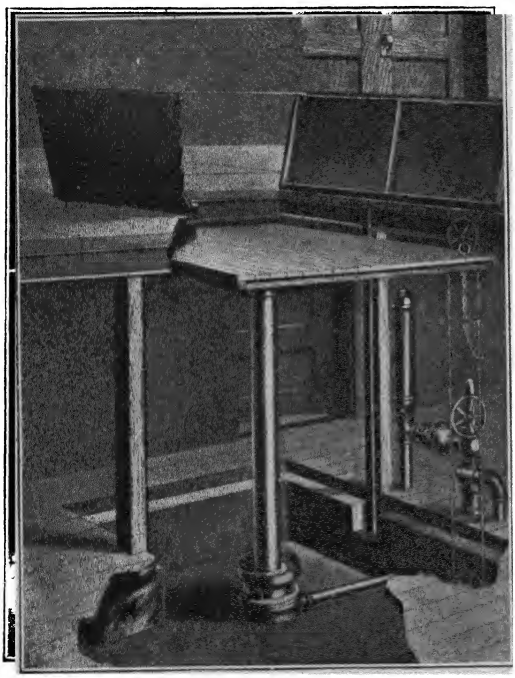


Fig. 106. Typical Plunger Hydraulic Sidewalk Elevator
*Courtesy of Warsaw Elevator Company, Warsaw, N. Y.*

The operating valve is located either at one side of the hatchway or in the pit. When the latter arrangement is used, the pit must be over three feet deep. The usual guide rails are provided for the

platform. When the travel of the car is more than one story, a counterpoise must be added to offset part of the weight of the plunger.

It will be seen from this description that this type of elevator is much simpler than either the horizontal or vertical hydraulic type previously described. However, since the introduction of the high-pressure system, the necessity that has arisen for various adjuncts and appliances has tended to make it quite complicated.

**Plunger.** For elevators of a low run, say from ten to twenty feet, the machine is very simple. The plunger in all cases is closed at its lower end and the water does not enter it at all, this being essential to its efficiency. For short plungers the lower end is closed by a screw cap, the projecting metal of the cap forming a stop to prevent the plunger from being forced out of the cylinder in case the operating cable should break or the usual limit stop become deranged. Sometimes the plunger bottom is grooved to relieve the pressure in such cases. The pit is usually supplied with bumpers on which the platform settles at the end of the down stroke, thus preventing the lower end of the ram or plunger from knocking out the bottom cylinder head.

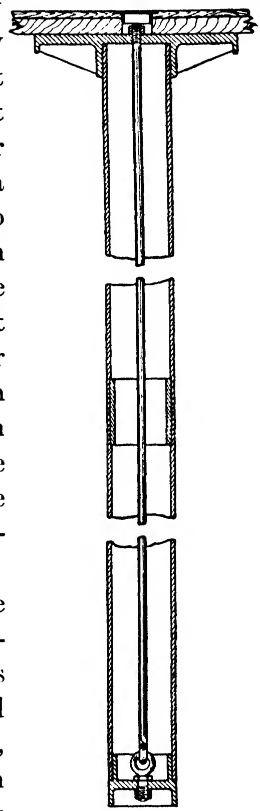


Fig. 107. Method of Fastening Sections of Plunger Together

*Cast-Iron Plunger.* When plungers were made of cast iron, screw joints were not generally depended upon. Usually the inner sides of the ends of sections were bored smooth and true for a distance of three or four inches, or, in the case of plungers of large diameter, even six inches, and a cast-iron ring or thimble was turned accurately to fit the bore. This thimble was made a little shorter than the combined bore of the two ends of adjacent sections. When the plunger was put together, a thimble was slipped in place inside the ends of the sections coming together, and a hydraulic cement made of a mixture of litharge or red lead and boiled linseed oil was used

on the joints. Where ends of the sections came together they were faced true, and through the entire plunger from top to bottom a long wrought-iron rod of $1\frac{1}{4}$ to $1\frac{1}{2}$ inches in diameter was passed. Such a rod, Fig. 107, had a thread at each end and, after the top and bottom plates closing the ends of the plunger were in place,

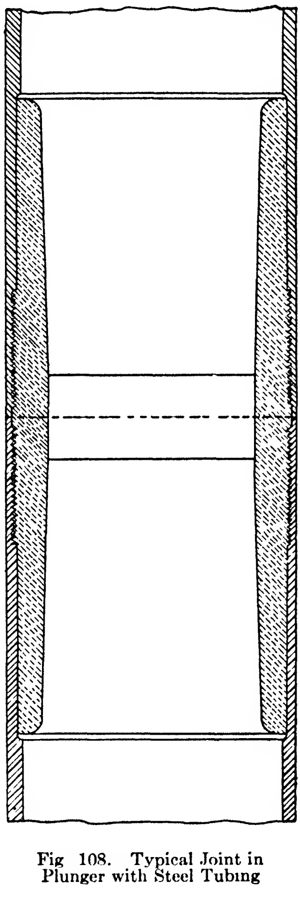


Fig 108. Typical Joint in Plunger with Steel Tubing

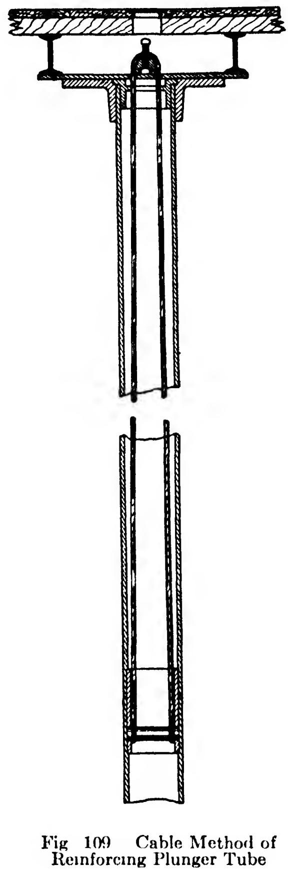


Fig 109 Cable Method of Reinforcing Plunger Tube

nuts were screwed tight on the ends of the rod, thus binding the whole together.

*Steel Plunger.* The introduction of steel tubing revolutionized this plan. The inside of the steel tubes was bored and threaded at the ends, the ends of the sections squared as before, and the thimble screwed to fit into the screwed ends of the plunger, as

shown in Fig. 108. In putting this plunger together the same hydraulic cement was used as with the cast-iron plunger. The whole was screwed tightly together, dependence for safety being placed on the high tensile strength of the steel of which the plunger was made. This could not be done with cast iron, because its tensile strength is very low and its crystalline nature makes it too brittle for the purpose.

*Long Plungers.* With the very long, and consequently very heavy, plungers in use at the present time a combination of both these methods is used. The plunger is made in sections, as previously described, with screwed ends and threaded thimbles, but in addition a wire cable is used inside, while a pin is set horizontally across the inside somewhat below the center, Fig. 109. At one end of the pin a wire cable is attached, the cable being led up to the top of the plunger and through the plate at the top and over a half-round block on the plate, thence down again inside the plunger to the other end of the pin, where the other end of the cable is made fast. Through this half-round block is a bolt or bolts, set in such a manner that by turning them the block may be raised and a tension secured on the cable to keep it tight.

By this means the weight of the lower part of the plunger is hanging on the couplings, this being a great relief to the couplings in the upper part of the plunger. Of course these strains on the coupling occur only momentarily as, for example, when the discharge valve is suddenly opened for lowering when the car is at the top of the run and the plunger is almost wholly out of the cylinder. With this arrangement the couplings below the center of the plunger are also subjected to this strain, but only to half the amount the upper coupling would have to carry, if the cable were not present.

**Cylinder.** The plunger does not touch the inside of the cylinder at all, a space of one inch or more being left all around the cap which closes the end of the plunger. The plunger runs through the stuffing box, and the packing therein keeps the water in the cylinder from escaping. At the upper part of the gland of the stuffing box, Fig. 110, is an annular space for grease, which is supplied from a compression grease cup. Above this is a recess for the reception of a wiper of soft rubber, which is kept in place by a ring attached by bolts to the top of the gland.

Fig. 111 shows with considerable detail the construction of the

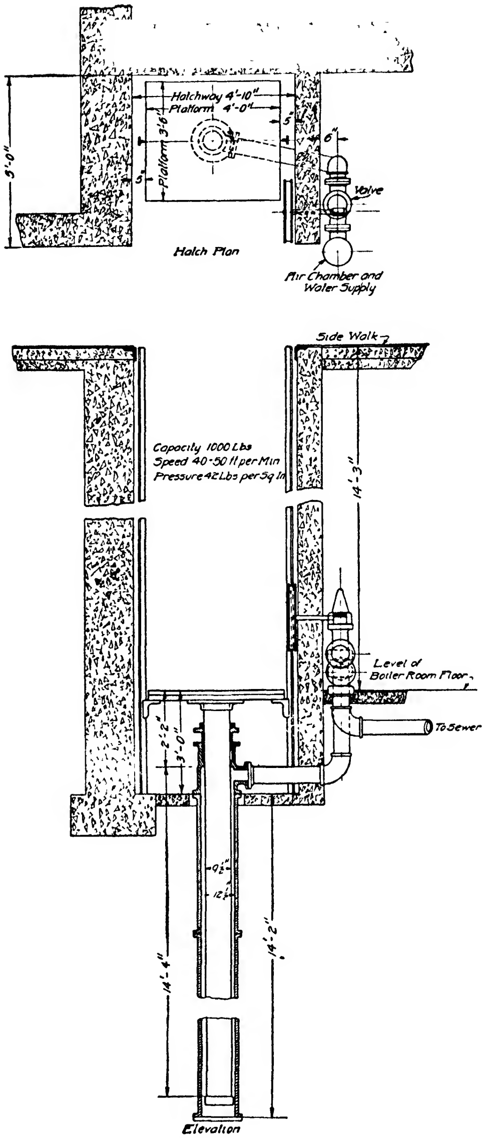



Fig. 110. Section of Plunger Hydraulic Elevator, Showing Details of Construction
*Courtesy of Kaestner and Hecht Company, Chicago*

cylinder and plunger of a passenger elevator. A hole is drilled into the ground slightly deeper than the total rise of the elevator. This hole is lined with casing until solid rock is encountered, the casing

being of heavy steel pipe. Where there is solid rock, steel casing is not needed.

The cylinder consists of heavy steel pipe screwed together with butt joints and welded at the bottom end. The top end is provided

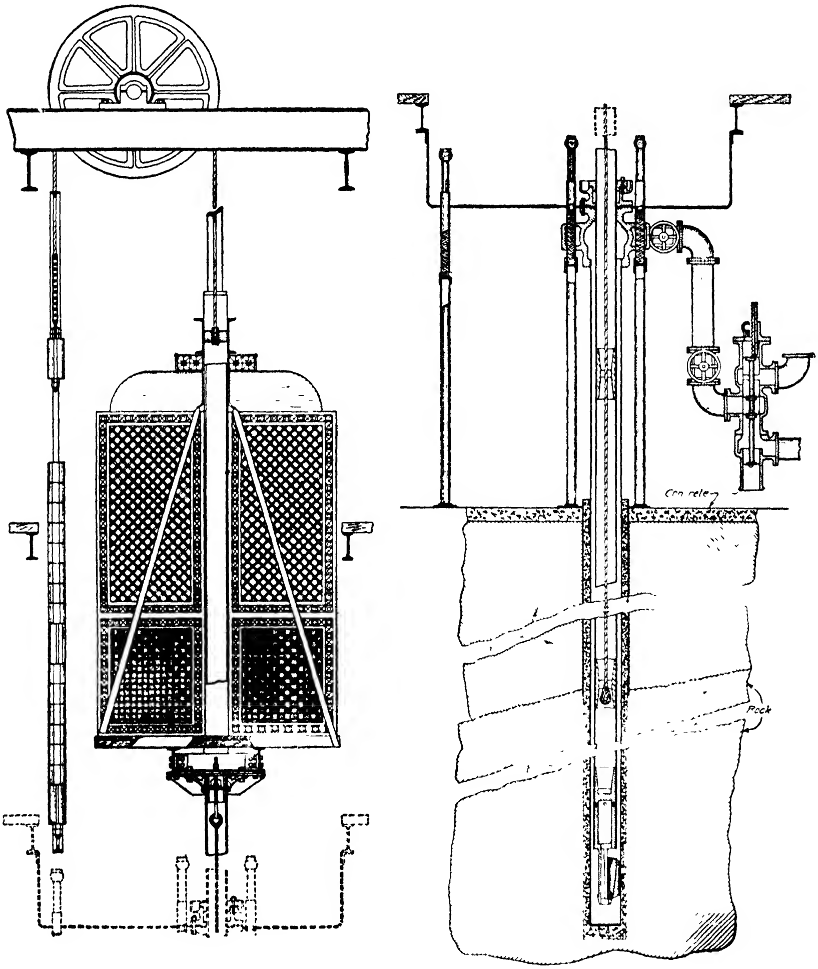



Fig. 111. Details of Cylinder and Plunger of Hydraulic Elevator
*Courtesy of Standard Plunger Elevator Company, Worcester, Massachusetts*

with a heavy casting known as the cylinder head, with a stuffing box through which the plunger passes, as previously stated. The cylinder is lowered into the hole and is usually provided with

a heavy plate which is used to hold the cylinder in position, the plate resting on the concrete at the top of the hole.

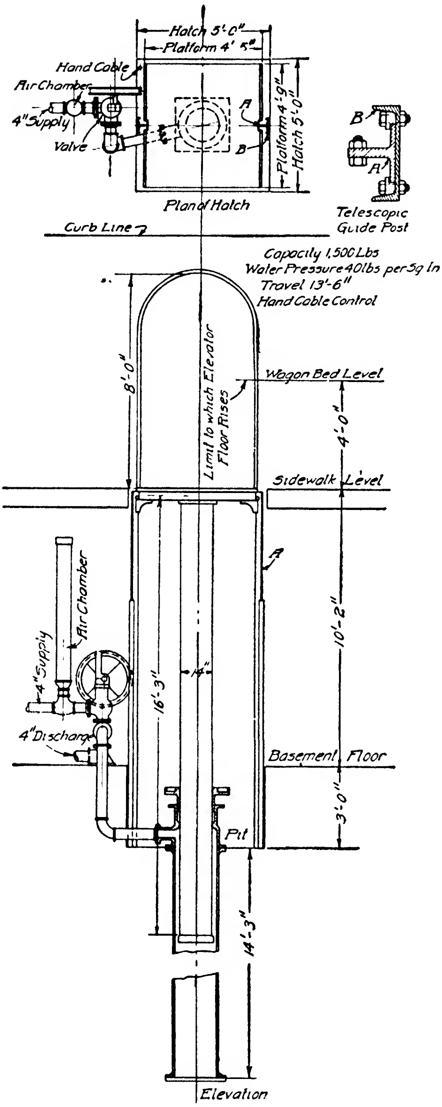



Fig. 112. Sidewalk Lift with Telescopic Guides

**Operating Chain and Ring.** In cases where the elevator runs up level with a sidewalk, street, or alley nothing can appear above the sidewalk, and hence the usual striker or stop arm of the platform or car has to be omitted. In its place a chain passing through a hole in the platform and having a ring at either end is used, as shown in Fig. 106. The operating cable, upon which stop buttons are set, runs through the ring at one end of the chain. The stop buttons are so arranged that, as the car approaches either end of its travel, the ring engages with one of them and pulls the cable in the direction to operate the control valve so as to stop the elevator. When the platform is level with the upper landing, the car may be lowered by pulling upward on the ring located on the car platform, thus opening the valve for descending. As the car approaches the lower limit of its travel, the ring passing over the operating cable engages with the lower stop button and shuts off the water.

**Telescopic Guide.** Sometimes where an elevator of this description runs up through a sidewalk near the curb, it is

found desirable to run the floor above the sidewalk to the level of a wagon bed. In such cases what is called a telescopic guide is used. Fig. 112 shows that this consists of guides sliding within rigidly fastened ones. Upon the arrival of the car at the level of the sidewalk the guide shoe on each side of the car comes in contact with a bolt set in the top of the inner guide and lifts it. This inner guide then slides up in the outer guide as far as the car travels. When the car descends on its return trip, the inner guide descends with the car to its original position, after which the guide shoes continue to travel on the inner guide, the outer guide being used merely as a support for the inner.

**Trapdoor Arch.** It frequently happens, also, that where the opening at the upper landing is covered by trapdoors, the elevator has to be so made that it will open these doors. In such cases a bow or arch, shown in Fig. 112, is attached to the car, which pushes the doors open as the car ascends. When the car descends the doors close, the arch preventing them from slamming.

**Counterpoise Weight.** In designing a machine of this type it is essential, therefore, to take into consideration, besides the load to be lifted, the force required to lift the plunger and car, and the inner or telescopic guides, and to open the doors. The area of the plunger must be so proportioned as to enable the machine to perform all these duties with the water pressure available. With elevators of this kind which run inside a building for a distance of from three to twelve stories, the weight of the plunger is a very important factor and the use of a counterpoise becomes imperative.

The plunger being hollow, has, when wholly immersed in the cylinder, a certain amount of buoyancy due to the weight of water it displaces, but as it rises out of the cylinder the buoyant effect gradually decreases.

The loads lifted in elevators of this type in tall buildings where the high-pressure system is in use vary from 2000 to 3000 pounds. Where the working pressure is about 500 pounds to the square inch the area of the plunger needs to be comparatively small and the plunger is made as light as is consistent with strength. Such plungers, as now made, vary in weight from 25 to 35 pounds per foot of length. Hence a 150-foot plunger will weigh somewhere about 4000 pounds. A car of about 6- by 6-foot floor area will weigh approxi-

mately 1800 pounds, and will carry enough people to weigh 2600 pounds. The total weight of the plunger, car, and people will be 8400 pounds. At a pressure of 500 pounds per square inch a six-inch plunger will do the work, allowing a good margin of reserve power to meet any emergency; but a plunger of this diameter when fully immersed displaces an amount of water equal in weight to 1800 pounds. Therefore this is the difference in lifting capacity of the plunger between the upper and lower landings. To equalize the lifting capacity as far as possible a counterpoise must be used.

*Weight of Counterpoise.* If the counterpoise were made as heavy as the plunger and car combined, the car would not descend except when loaded to its full capacity. The counterpoise weight must not approach the weight of the plunger and cab by at least the weight of the water displaced, and a few hundred pounds should be allowed for overcoming friction, etc., in descent; but even with this arrangement, the machine will lift 1800 pounds more at the lower landing than it will at the top. This difference in lifting power at different elevations is compensated for in the following manner.

*Effect of Weight Cables.* The counterpoise weight is hung by, say, about three times as many cables as would be ample in strength to carry the weight, the additional cables being used to offset as far as possible the difference in lifting power of the machine at various levels. These cables, which are ordinarily of $\frac{5}{8}$-inch diameter, weigh about $\frac{3}{4}$ pound to the lineal foot, or for the six, $4\frac{1}{2}$ pounds per foot. Now, when the car is at the top of the run, all six of these cables would be on the opposite side of the sheave over which they run, or, in other words, they would be hanging in the weight run helping the weight to pull up on the plunger. The combined weight of these six cables, 150 feet in length, would be 675 pounds. When the plunger is at its lowest point, these cables hang down in the hatchway, pulling up on the counterpoise weight with a force of 675 pounds, thus helping the plunger to displace the water in the cylinder. It will be readily seen that the total influence of these cables on the lifting capacity of the plunger would be 1350 pounds, for when the car is at the top of the shaft 675 pounds will be added to the counterpoise weight, while when the car is at the bottom this amount will be subtracted. Hence the difference in lifting capacity between the top and bottom landings is only 450 pounds instead of 1800 pounds.

These figures and dimensions have been given merely to illustrate the point desired, for, in actual practice, such pressure of water for the height and load named would be greater than necessary. With the counterpoise attachment and pressure given, a six-inch plunger would be larger than necessary. A four-inch plunger would be of sufficient area, but a four-inch plunger for such a height would be likely to buckle. A lower working pressure and larger ram would therefore be more practical.

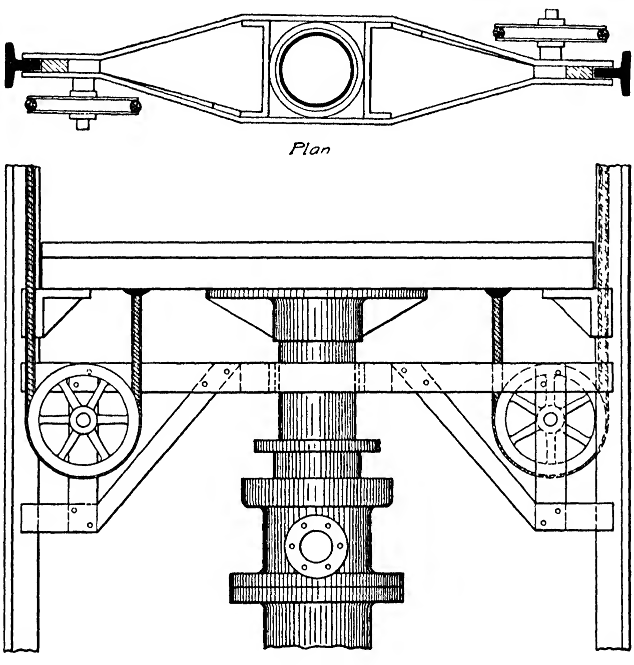



Fig. 113. Hydraulic Elevator Plunger with Traveling Stay

**Traveling Stay.** Even when the plunger is well proportioned, there is a tendency for it to buckle under heavy loads and great heights. To prevent this an ingenious device, shown in Fig. 113 and known as a traveling stay, is used. A ring is bored to fit the plunger loosely, and is fastened to a frame which extends on each side to the guides on which the car travels. To the ends of this frame are fitted two guide shoes to travel on the guides, and two sheaves, one on each side and revolving on pins attached to the arms. Two ½-inch

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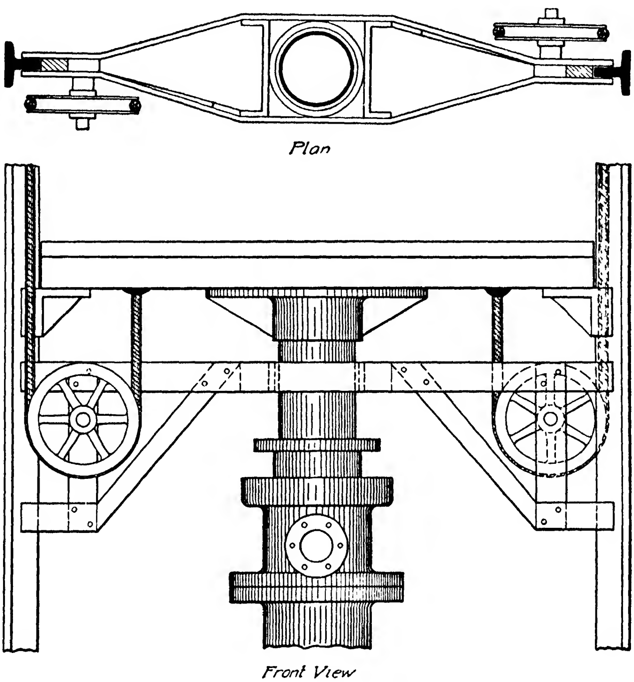


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cables, one on each side, are led from a point a little more than halfway up the run, down the hatchway, and under each sheave, thence up to the bottom of the platform, where they are made fast.

With this arrangement, the car on ascending will carry this yoke or frame up with it, but, due to the sheaves, the frame will travel only half as far and half as fast as the car does. Hence, when the car has reached the top of its run the yoke will be halfway up the shaft, or at the point where the plunger would be inclined to bend out of a straight line. Hence if there is a tendency to bend, the frame will prevent it. When the car descends, the frame will also descend of its own weight.

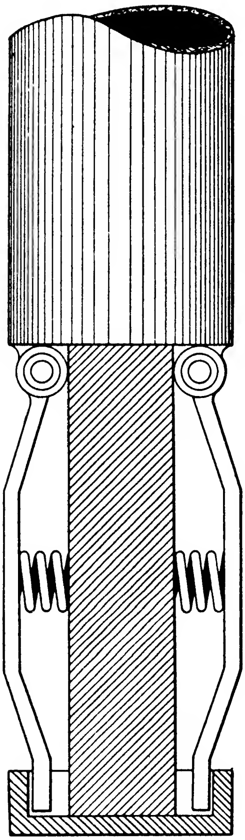


Fig. 114 Spring Type of Plunger Stay

**Plunger Stays.** In the case of a plunger for a run of one hundred feet or more, the pressure of the water against its lower end, when the car is at the bottom level, has the same tendency to buckle it as when the car is at the top of the run, for it must be remembered the cylinder is not bored smooth and the plunger is supported only at the stuffing box.

*Plunger Shoes.* The lower end of the plunger is often fitted with three or four shoes, Figs. 114 and 115, which rub on the sides of the cylinder. These shoes are made rather long and their ends rounded so that they will freely pass any minor inequalities or roughness on the inside of the cylinder. To further assist in their movement they are fitted with springs to aid them in overcoming obstacles.

*Plunger Brushes.* Instead of bronze shoes some prefer to use stiff brushes with bristles of hard-brass spring wire set in a soft metal back, which is set in a groove in the bottom end of the plunger. Three or four of these brushes are used, equally divided in space around the plunger, but in such cases the spring used in the other type to force the shoe out against the sides of the cylinder is

omitted and the soft metal back, which is usually babbitt, is held rigidly in place. The spring-wire bristles are longer than is necessary to reach the sides of the cylinder, and hence act as springs until they are worn too short to be of further service, when new brushes must be put in. Both devices work well, but the shoes are probably the more substantial.

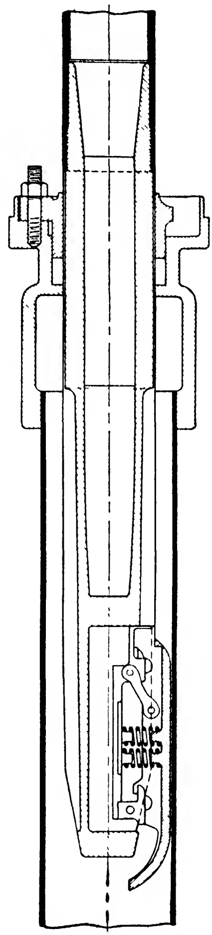


Fig. 115. Shoe Type of Plunger Stay

**Limit Devices.** *Limit Valve.* With such an arrangement as here described, it has been found necessary to use limit stop or automatic stop valves, as shown in Figs. 116 and 117. These are somewhat similar to those described as being in use with the horizontal and vertical hydraulic machines, but the mechanism of this machine will not admit of a similar arrangement for their operation. In this case two limit valves are used, one for each end of the run. Both are situated in the pit near the operating valve and close to the top of the cylinder. To operate them, two cables are used, one for each valve. These are ½-inch tiller cables and extend from top to bottom of the run. The upper ends of these cables are fastened permanently at the ceiling or to the beams carrying the counterpoise sheaves.

Both limit valves are arranged with their operating sheaves at the same side of the hatchway, and the cable for the lower limit valve is led down the opposite side of the hatchway to that on which these sheaves are located, and thence diagonally across beneath the car to the valve sheave. A sheave is then attached to the bottom of the platform in which the cable will run, the latter being kept taut so that it will remain in the groove of the sheave. When the car approaches the lower landing, it carries the cable out of straight line, thus putting a strain upon it which causes the cable to unreel from the valve sheave and operate the valve.

The cable attached to the sheave of the upper limit valve is run up the hatchway on the same side the sheave is located, but at the

top it is attached at the opposite side alongside the other cable. A sheave is attached to the top of the car to run on this cable, and when the car approaches the upper landing, a similar movement of the upper limit valve is produced in a like manner. Each valve is supplied with a lever and weight to bring it back to its "open" position upon the return of the car. This description of the arrangement of the limit stop valve is somewhat general, but all limit valves work on approximately the same principle.

*Hollow-Plunger Device.* Some makers, besides using the limit valves on the high runs, use another device to prevent the possibility

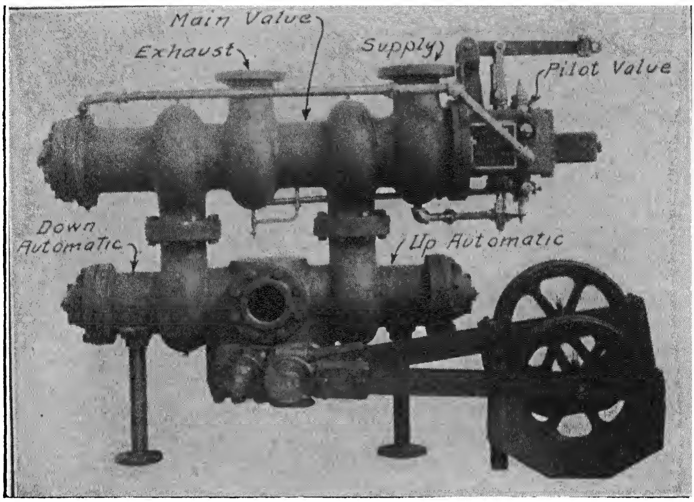



Fig. 116. Side View of Limit Stop Valve for Direct-Plunger Type
*Courtesy of Standard Plunger Elevator Company, Worcester, Massachusetts*

of running the platform so high as to jam the car up against the overhead beams, should the limit stop valves fail. This scheme is to attach to the lower end of the plunger, by means of a strong bolt, another section of plunger which is hollow. It is attached below the closed end of the plunger, and carries the shoes or brushes for centering. This section is made of brass, so that it will not corrode and thus become so thick with rust that, when required to act, it will pass the stuffing box.

This section being hollow and open at its lower end, the water has free access to all parts of it, and near its upper end, where it joins the plunger proper, there are three or four holes through its sides

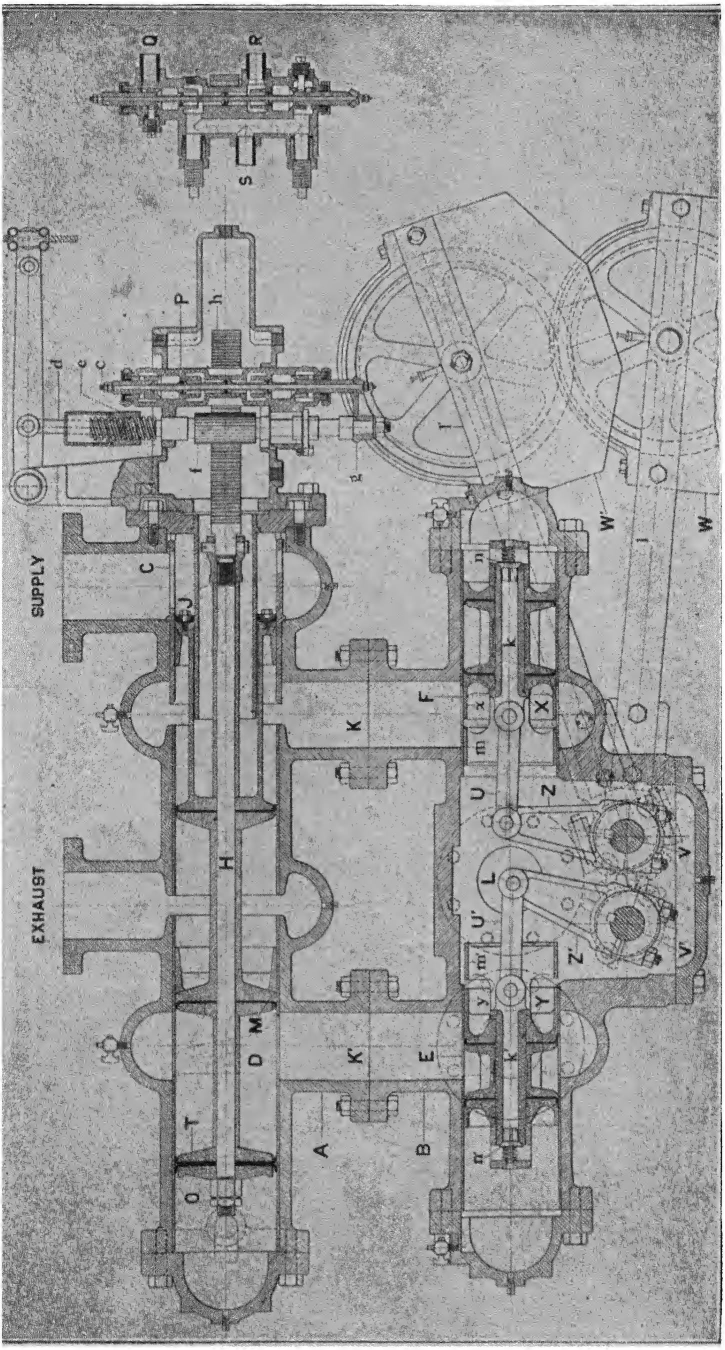



Fig 117 Sectional Details of Valve Shown in Fig. 116
*Courtesy of Standard Plunger Elevator Company, Worcester, Massachusetts*

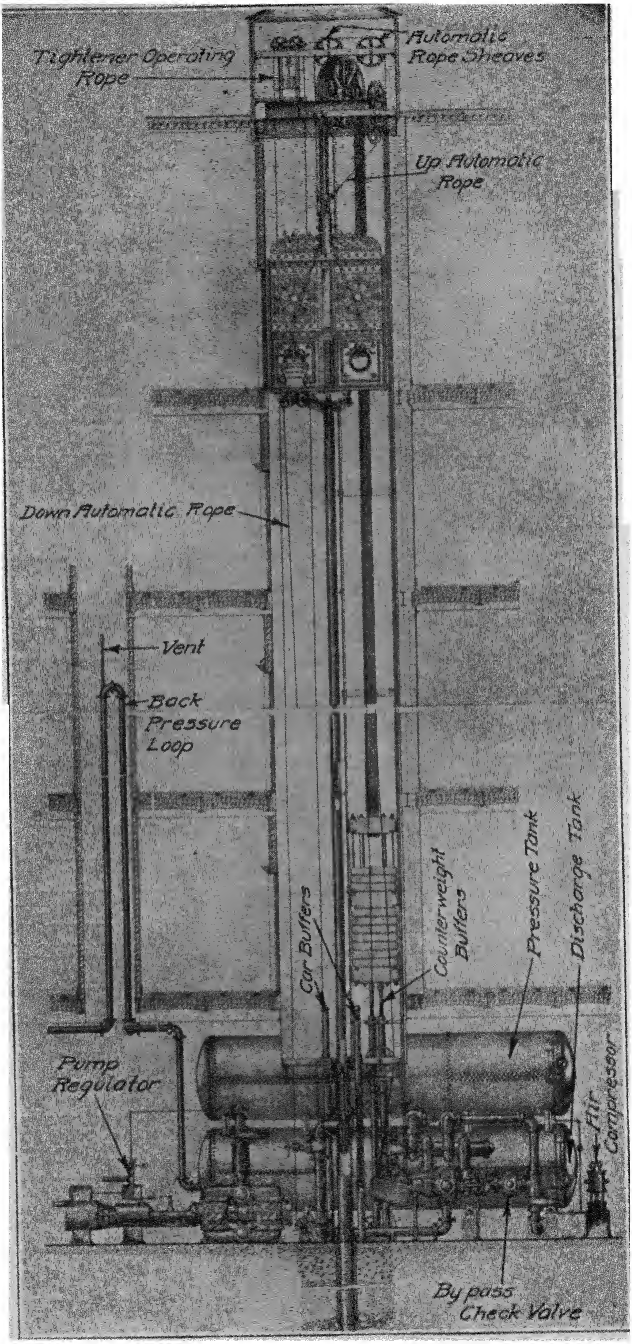



**Fig. 118. Vertical Section of Typical Back-Pressure Loop Elevator Installation**

***Courtesy of Standard Plunger Elevator Company, Worcester, Massachusetts***

which are equal in area to the feed pipe. Should the limit stop valve fail to work and the plunger continue in its upstroke until the platform should rise, say 18 inches above the top landing, this lower brass section of the plunger would then pass on up through the packing in the stuffing box until its holes would be above the gland. Then, of course, any further water fed to the cylinder would escape through these holes into the basement; but this is a lesser evil than jamming the car into the upper beams. When this device is used, both the cylinder and plunger are made longer than for the ordinary arrangement.

**Operating Valve.** In some minor details the construction of the operating valve is slightly different from the valve used with the hydraulic elevators before described, but in the essential points it is similar, the slight differences being due largely to the individual ideas of the various makers. The means of operation is identical with that of the other types, namely, by hand cable, lever pilot wheel, and pilot valve, according to the pressure, speed, distance traveled, and the service for which the elevator is designed. Many of these elevators are operated by the push-button control previously described, and the same objections exist with this type of machine as with the others.

**Back-Pressure System.** Fig. 118 shows a complete back-pressure loop elevator installation. The objects of the back pressure are to assist the pump, to act as a water counterbalance for the descending car, and to prevent space forming underneath the plunger. Two closed tanks are shown in the figure. The upper tank is the pressure tank containing water under high pressure; the lower tank is the discharge tank, and is charged with low pressure. The high pressure is caused by compressed air generated by the air pump. The low pressure is caused by the back pressure in the loop or vertical water column. This amounts to 2.3 pounds per square inch per foot of rise.

In the case of a plunger elevator ascending at high speed, if the valve should close too suddenly, the plunger might leave the water and a disagreeable jar would be the result as the plunger dropped back. To prevent such condition a by-pass is provided from the low-pressure discharge tank to the elevator cylinder, a check valve being in the line, and in the event of the plunger leaving the

water the check valve is opened, allowing the water from the low-pressure tank to fill in any space that might occur as above described.

At the top of the back-pressure loop is a vent to prevent the water siphoning. The lower end of this pipe is connected to the sewer, and, in the event of the discharge tank becoming overcharged, the water would simply flow from the loop and into the sewer.

*Operation.* The cycle of operation of the tank system as shown in the illustration is as follows: The pressure tank contains about $\frac{2}{3}$ water and $\frac{1}{3}$ compressed air. When the valve is opened to raise the elevator, the compressed air forces the water through the valve to the elevator cylinder, causing the elevator to rise. When the valve is shut off, the elevator rests upon a column of water in the cylinder. When the valve is opened to lower the car, the car descends by gravity, forcing the water out of the cylinder into the low-pressure tank.

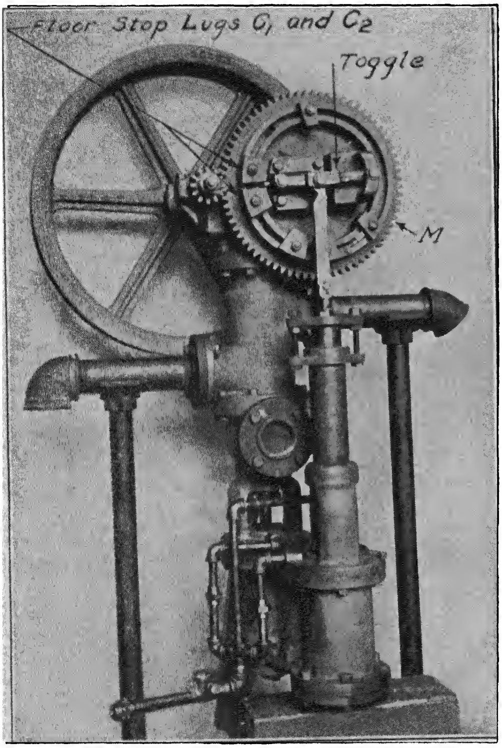



Fig. 119. Pilot Valve Designed to be Electrically Controlled from Car
*Courtesy of Standard Plunger Elevator Company, Worcester, Massachusetts*

The steam pump shown in the illustration is provided with a regulator governed by compressed air. When the pressure reaches the maximum point, the steam throttle is closed automatically and, likewise, opens as the pressure falls. Therefore, the pump automatically takes the water from the discharge tank and forces it into the pressure tank as fast as it is exhausted from the elevator cylinder. The air compressor is used only occasionally to make up for any slight loss caused by leakage.

**Electric Control.** An automatic valve designed to be electrically controlled from the car is shown in Fig. 119, and the general arrangement of the valve and controller is shown in Fig. 120. The valve is governed in its opening movement by electrically operated pilot valves and is closed by the movement of the car itself, thus insuring absolutely accurate stops.

The valve is of the regular three-way balanced rack-and-pinion type, except that a motor cylinder is used at the lower end for opening the valve and another motor cylinder is used to operate a clutch to stop the car. The valve is closed by means of a running rope, which is attached to the car and connected to the valve drum as previously described. As the car moves the drum revolves, a clutching device being used to center the valve.

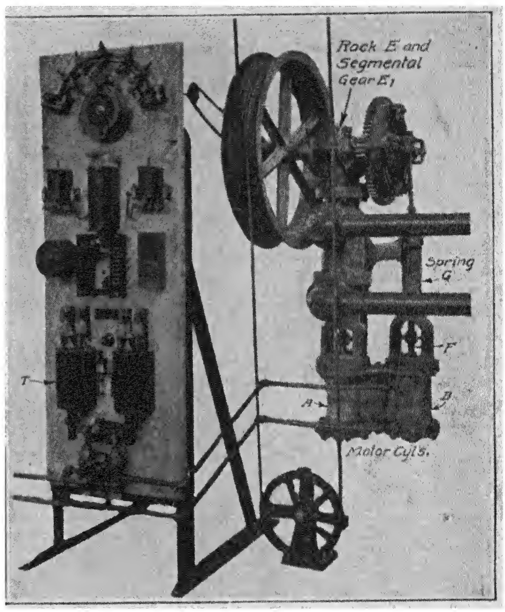


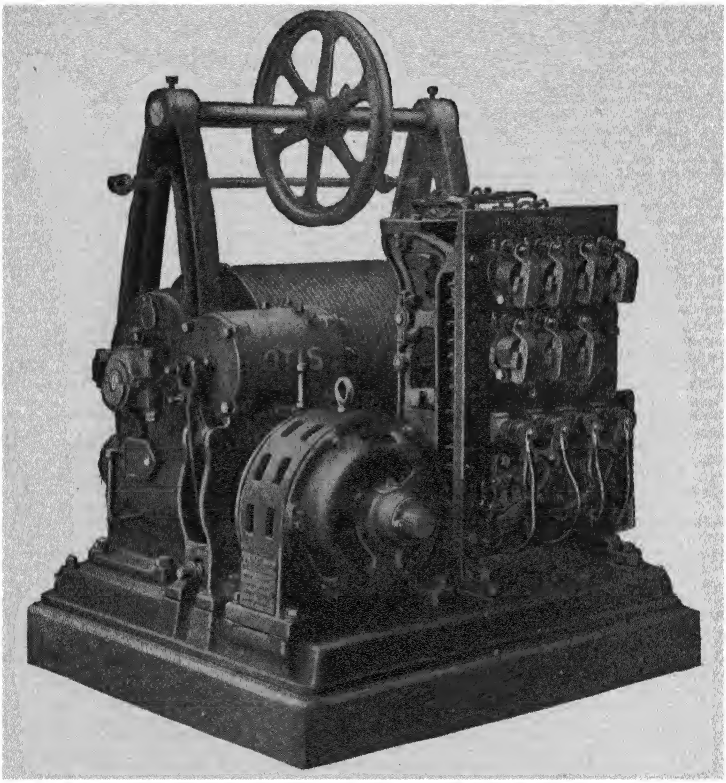

Fig. 120. Arrangement of Valve and Electrical Control
*Courtesy of Standard Plunger Elevator Company, Worcester, Massachusetts*

The rack $E$, Fig. 120, at upper end of valve stem engages a segmental gear $E_1$, which is direct-connected to a clutching arrangement. When the elevator is started, the rack $E$ is moved up or down, as the case may be, by motor cylinder $A$, and the clutching arrangement is set in an angular position and must be returned to a horizontal position when the valve closes. The clutch is opened by pressure in the motor cylinder $B$ and closed by the coil spring $G$. When the pilot valve is opened to run the car up, water is admitted to the motor cylinder, forcing the piston $F$ down and uncovering the port to the motor cylinder, $A$. The water passing into motor cylinder $A$ forces the piston down and opens the main valve.

The pressure in motor cylinder $B$ overcomes the spring $G$ and

releases a clutch bar which is between the two floor-stop lugs $C_1$ and $C_2$, there being two lugs for each floor, See Fig. 119. The clutch bar is moved back by means of the toggle connections, and the piston in motor cylinder $A$, when moving, shifts the clutch in an angular position by means of the rack $E$ and segment gear $E_1$. When the elevator is running at full speed, the pressure holds the two pistons in the motor cylinder down or up, as the case may be, and when this pressure is released by the pilot valve closing, the spring $G$ instantly returns the piston $F$ to a central position, and the clutch bar engages the slot between the floor-stop lugs, similar to $C_1$ and $C_2$. The continued movement of the drum, which is driven by the rope connection to the moving car, revolves the clutch back to a horizontal position, incidentally centering the main valve stem, which is connected through the rack and segment gear. It should be noted that the gear $M$ has two floor-stop lugs for each floor, and, as the gear revolves, the lugs representing the floors pass the clutch in regular order until the clutch acts.

The pilot valve which governs the main valve is unbalanced to insure positive seating and is solenoid-controlled, as shown at $T$ in Fig. 120. A dashpot cushion is provided at the lower end of stem to prevent the unbalanced valve jarring when seating. The stem simply drops into a pocket of water, which slightly retards the quick-moving stem.




**OTIS MULTIPHASE ALTERNATING-CURRENT RESIDENCE ELEVATOR MACHINE WITH FULL AUTOMATIC CONTROL**

*Courtesy of Otis Elevator Company*

# ELEVATORS

## PART III

## ELECTRIC ELEVATORS

**Early Types.** The first application of electricity as a motive power for elevator service was made in the year 1889, and was doubtless suggested by its successful use on street cars and in driving lines of shafting in industrial establishments. The installation consisted of an ordinary shunt-wound motor belted to a countershaft, which in turn drove a two-belted worm-geared hoisting apparatus. The starting, stopping, and reversing movements of the cage, or platform, were obtained theough the medium of an operating cable by means of which the open or cross belt was shifted as desired from the loose pulley to the driving pulley of the hoisting machine or *vice versa;* a brake was applied automatically with the cable at stopping position.

In this type of machine, the motor had to be started and allowed to attain normal speed before the elevator could be put into service and, furthermore, the motor had to run continuously even when the elevator was being loaded or unloaded. This arrangement was very wasteful of electric current and the speed was limited.

Various attempts were made to overcome these difficulties, and in many cases the methods used were unique and proved impractical. One of these, the Pratt-Sprague, consisted of a long screw running horizontally in bearings at either end, which were driven directly by a motor placed at one end. The screw ran in a nut having a cross head, which traveled horizontally on guides, the same as the cross head of a horizontal hydraulic elevator, and was supplied with sheaves on either end. The construction of the machine was such that a double set of traveling sheaves and also fixed sheaves were necessary. The cables were rove over these sheaves like any horizontal hydraulic, and the motor was reversible.

One of the principal features of this type of machine was the construction of the nut which traveled on this large screw. It was

supplied with steel balls on the "pull" side of the screw, and these ran close together in single file through a channel, which carried them around through the threads of the nut and caused them to return to the other end of the same after they had passed through. Of course, there had to be so many of them that they completely filled the channel from one end to the other, and it was thought that their use would reduce the friction to a minimum. It was found, however, in practice that flat spots were soon worn on them and they ceased revolving, thus cutting grooves or scores in the thread of the screw —a very serious matter. Elevators of this type were also very prone to become deranged, and their operation was not as economical as had been anticipated.

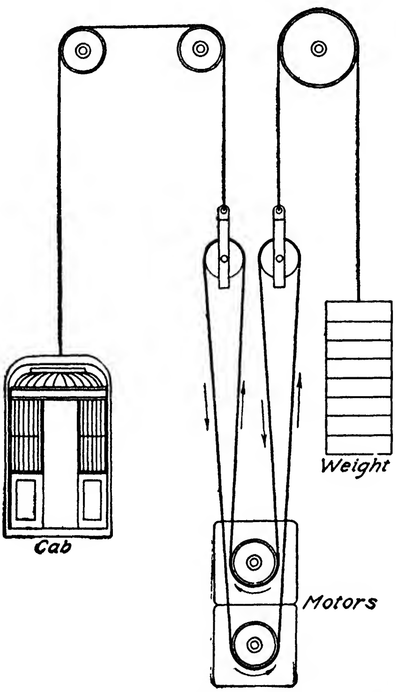


Fig. 121. Diagram of Fraser Elevator Driving Mechanism

The controlling device was quite novel and the operation of the cage very agreeable and pleasant. The control of the motor driving this screw was effected by means of a small pilot motor, which in turn was operated by means of push buttons in the cage. Very few, if any, of these elevators are in use at the present time.

Another type was the Fraser elevator, the driving mechanism of which consisted of two motors set one above the other. These motors were necessarily of the slow speed type, running about 420 r. p. m., and each one had upon the armature shaft sheaves of about 20 inches diameter. Cables were arranged so as to form a double bight or loop around each of these pulleys on the armature shafts, as in Fig. 121. The cables ran from these sheaves over two pulleys, one attached to the car cable, and the other to the counterpoise cable. The motors ran in opposite directions, and by means of rheostats placed in the cab the fields were weakened or strengthened, in

order to vary the speeds of the motors. Referring to the diagram it may be seen that when both motors were running at the same speed, no motion of the car would result; but by varying the speed of either motor, the car would run at a speed equal to half the difference in speed of the two motors. No motor reversing apparatus was needed, the motion of the car being obtained entirely by the change in speed; this gave a most desirable method of making stops and starts. The machine was very severe on the cables, so destructive in fact that they had to be frequently renewed; and taking it altogether, the design was not found so desirable as had been anticipated, from the standpoint either of economy or maintenance. However, actual results in operation were all that could be desired, including the speed attained and smoothness of stops and starts.

**Difficulties with Variable Speeds.** In the latter part of 1889, the first attempt was made to run elevators by means of the motor attached directly to the worm shaft. Two elevators built on this principle were installed in buildings on Broadway, New York City, but they were not eminently successful, owing to their inability to start under a full load and to difficulties experienced with the controllers.

It was soon discovered that a motor which was entirely shunt-wound did not have sufficient torque to start the load unless a motor enormously large for the work was used, thus entailing an additional expense which did not seem warranted. A trial of a series winding in addition to the regular shunt winding was found to solve the problem satisfactorily as far as starting was concerned, but the use of the compound motor caused the elevator to have a variable speed depending upon the load lifted. This speed variation interfered seriously with the stopping of the cage exactly at the different landings, even when the stops were made automatic by means of a striker attached to the cage, an arrangement which was quite popular and in general use at that time.

To remedy this defect the controller was so constructed as to cut out the series winding as soon as the motor had attained its normal speed, leaving it to run on the shunt winding only. This arrangement produced the desired results, viz, a normal and regular car speed under all loads and consequently perfect control in stopping the car at the various landings. The details of the method by which

this has been accomplished will be fully described under the heading "Control".

These difficulties overcome, the electric elevator became a real competitor of the older and better known hydraulic types.

## MOTOR DESIGN

**Requirements.** The motor requirements for elevator service differ materially from those for almost any other kind of work and the duties performed are, with possibly one or two exceptions, the most exacting.

In most lines of work the motor is started running light, and the load is applied after the normal speed has been attained. The elevator motor, however, must not only be able to start under full load, but it must pick up, or attain, its proper speed under that load and do it within a period of from four to eight seconds. After normal speed has been acquired, the speed must remain unchanged until the circuit is broken for the purpose of stopping.

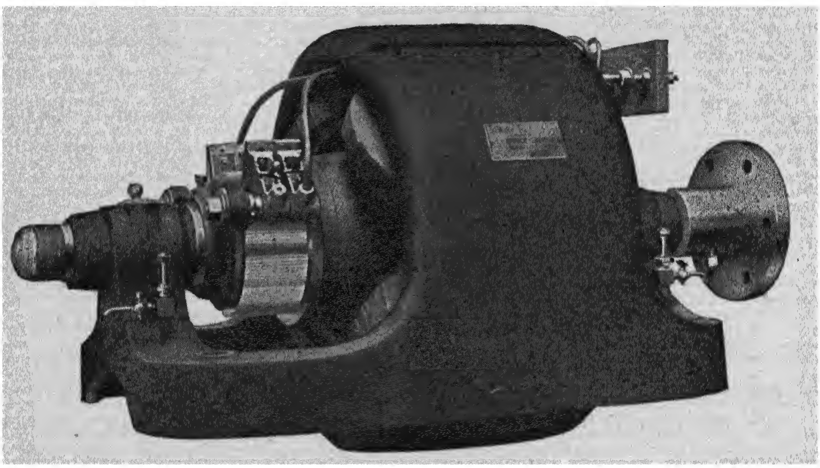


**Fig. 122. Direct-Current Elevator Motor**
*Courtesy of Warner Elevator Manufacturing Company*

*Direct-Current Motors Most Suitable.* To any one conversant with the subject, it will be apparent that a special design is absolutely essential for this particular service. With an alternating-current motor, only an approximate fulfillment of the requirements is possible; but with a direct-current motor, the degree of perfection is

indeed gratifying when compared with the service obtained only a few years ago. Some of the special features of the direct-current motor designed for this work are as follows: low speed; high starting torque; massive frame, Fig. 122, and field cores; ample shunt windings fully up to the full power of the motor; series windings equal to at least 50 per cent of the horsepower of the motor, these to be in sections so they may be cut out gradually as the motor picks up speed, and to consist of wire almost, if not quite, as large in carrying capacity as the service wires; brushes of sufficient carrying capacity and with an ample area of commutator contact; and rigid brush holders that will not jump or clatter when the motor is running.

*High Cost Inevitable.* A motor built on the lines just suggested and operated by a well-designed controller will give very satisfactory service and will last for years if kept properly cleaned and lubricated. This last condition is seldom met, for, generally speaking, elevator motors do not receive the attention they should have. Furthermore, the motors are not always of a design suitable for the work expected of them, and it has taken an enormous amount of urging, of argument, and of pleading on the part of the elevator maker to get the motor builder to recognize the peculiar needs of this kind of service. The motor manufacturer is not altogether to blame for this, for frequently the keen competition met with in securing desirable contracts has tempted the elevator builder to use an inferior motor—a practice which has invariably led to trouble. All attempts to produce a less expensive motor which would at the same time give the proper service have failed. Today the features enumerated above are fully recognized by all the best and most reliable motor makers as being absolutely essential to a serviceable and long lived motor for this kind of work.

**Faults in Design.** *Light Shafts.* One of the mistakes which designers often make is the result of early practice in connection with belted transmission. In these early forms, where the pulley of the motor was much smaller than the driven pulley and the motor started before the load was applied, the low starting torque allowed of a much lighter armature shaft than would be necessary when the motor started under full load. As a result of the change to direct-connected design and to the practice of picking up under load, many instances of shafts twisting off under the heavy starting torque

bear witness as to imperfect design or to an attempt to skimp material.

*Heating of Motors.* Again, motor designers in many cases reasoned that because a motor in this kind of service ran intermittently, it did not have time to heat to a dangerous extent before it was stopped, and during the interval before starting again it would cool. With this fallacious deduction as a basis, they built every part of the machine of light weight for the sake of economy in production, with the result that the motor was heated to a very unsafe degree in its efforts to start under load and the insulation around the copper wires comprising the windings of both armature and field was ultimately destroyed. The rewinding of these coils is an expensive operation and, of course, the loss of efficiency of the motor during the transition from perfect insulation to an actual short circuit in the coils is a more insidious and baffling process.

This brief description clearly explains the absolute necessity of having motors ample in power and of proper design, with liberal windings, field cores, and frame. Another practice which resulted in bad heating was that of keeping the field windings of the bipolar motors excited all of the time in order to produce a fairly prompt start. This resulted frequently in incipient fires and even where this did not occur it brought about the premature destruction of the field windings. Today, in all cities where regulations are in force to insure proper installation of electric elevator equipments, the current must be entirely cut off from the motor when the elevator is stopped, a regulation which has proved as beneficial to the elevator maker as to the owner of the plant. Furthermore, bipolar motors have been entirely discarded for this service by the best makers as they have been found less efficient than other types, especially in starting under load.

*Slow Speed.* It used to be contended that a slow speed motor was not so efficient as those of higher speed, but as the question of design has been more thoroughly investigated and understood, these objections have all been overcome. In fact, such a degree of perfection in the design of motors for elevator service has been attained that within the last few years motors running efficiently at a speed as low as 40 r. p. m. are frequently met with in connection with the so-called gearless type of elevator engine.

## MODERN TYPES

**Interpole Motors.** For variable speed elevators a type called the "Interpolar Motor" is now in general use. This motor, as its name would indicate, is a direct-current motor with additional poles and field windings interposed between the regular fields, making it an 8-pole motor, Fig. 123. The interpoles are provided with series windings, the wire used being as large in cross section as the line

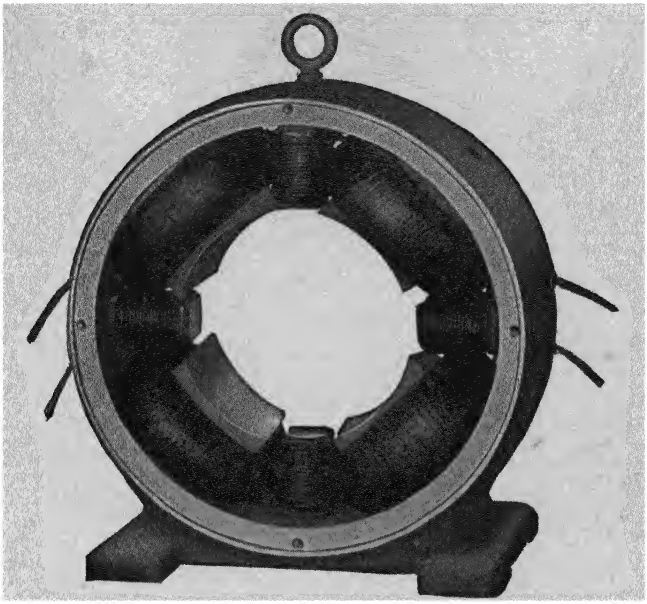


Fig. 123. Interpole Motor Field Ring and Coil
*Courtesy of Electro-Dynamic Company*

which feeds the motor and consisting of one continuous wire, one end of which is connected with the line and the other end with one of the armature leads. Current flowing in these windings energizes the interpoles, thus making them alternately north and south poles, or the reverse, according to the direction of the current passing through them.

The regular fields of the motor are wound both series and shunt, as shown on page 165. When the motor is started, the current flows through these heavy series windings and gives the motor abnormally heavy fields at the slow speed, thus enabling it to start under as heavy a load as it will be called upon to carry at full speed. The weakening of these abnormally heavy fields by decreasing the current

in the interpole windings and cutting out the series windings on the regular fields increases the speed of the motor to double its starting speed. This, of course, is done through the agency of the controller in one, two, or three steps, thus giving the motor two or three speeds as may be desired. These speeds are about 375 r. p. m. to start with, and 950 r. p. m. for full speed.

In high speed elevators running at 350 to 400 f. p. m. at maximum, this is a valuable feature in a motor because it enables the operator to start and pick up speed gradually, the acceleration occurring during a period of time covering only a few seconds.

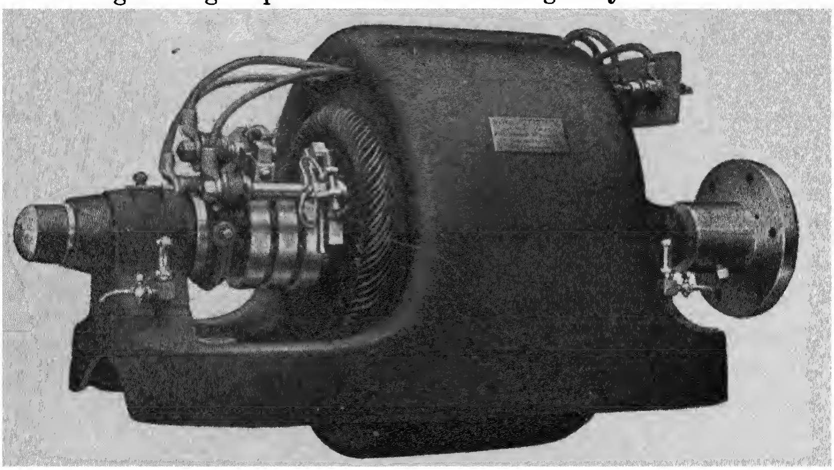


Fig. 124. Alternating-Current Type of Elevator Motor
*Courtesy of Warner Elevator Manufacturing Company*

These results may be obtained either through the medium of the operating switch or by an automatically operating device attached to the controller.

**Alternating-Current Motors.** With motors built to run on alternating current, Fig. 124, none of the features described are obtainable, the motor being simply started and allowed to attain its normal speed if the elevator is unloaded, or its nearest approximate when loaded. The variation between the synchronal and full load speeds is from 5 to 10 per cent, the alternating-current motor resembling to some extent the compound-wound direct-current motor in this respect. The greatest difficulty experienced with motors of the alternating-current type is their lack of ability to start under a heavy load, and for this reason proportionately larger sizes must be used,

the increase in horsepower required being fully 33 per cent. This increased size of the motor is really an advantage, for, besides giving a heavier torque at starting, it furnishes an excess of power which enables the motor to run at full speed without such noticeable fluctuations with changes in load as would be the case with a smaller motor.

## CONTROL

**Functions of Control.** Starting, accelerating, slowing down, stopping, and holding the elevator securely at the various landings are governed by the *control,* which is, therefore, a very important part of the operating mechanism of the electric elevator.

*Starting.* In starting, the brake is released and simultaneously the current is admitted to the motor. With direct-current motors, however, the full strength of the current may not be admitted until the motor has attained nearly its full speed. The current is controlled by passing it through resistance coils or through cast-iron grids made from iron of a known degree of purity and density. These coils or grids, which are arranged in sections, or *banks* as they are technically termed, are introduced into the circuit when the current is first admitted to the motor and by their resistance tend to cut the current down to a limit that is safe for momentary admission to the motor windings without endangering the insulation.

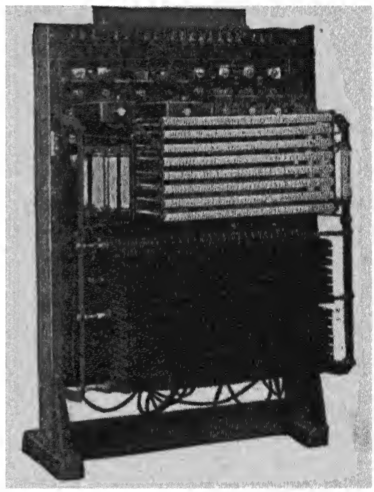


Fig. 125. Back View of Control Board
*Courtesy of Otis Elevator Company*

*Acceleration.* The moment the armature starts to revolve, the strength of the current flowing through it may be slightly increased without danger. As the strength of the current increases so does the speed of the armature, and with each increase of speed other sections of resistance are successively cut out of the circuit until the armature has attained nearly its normal speed. At this time the current is allowed to pass directly from the line into the armature, thus giving it the acceleration necessary for its full power and normal speed.

## TYPES OF CONTROLLERS

**Functions.** A controller differs from the ordinary motor starter in that it performs several other functions besides that of simply starting the motor. It controls the energizing or cutting out of the circuit of the solenoid which operates the brake; it reverses the direction of rotation of the armature as desired; it controls the resist-

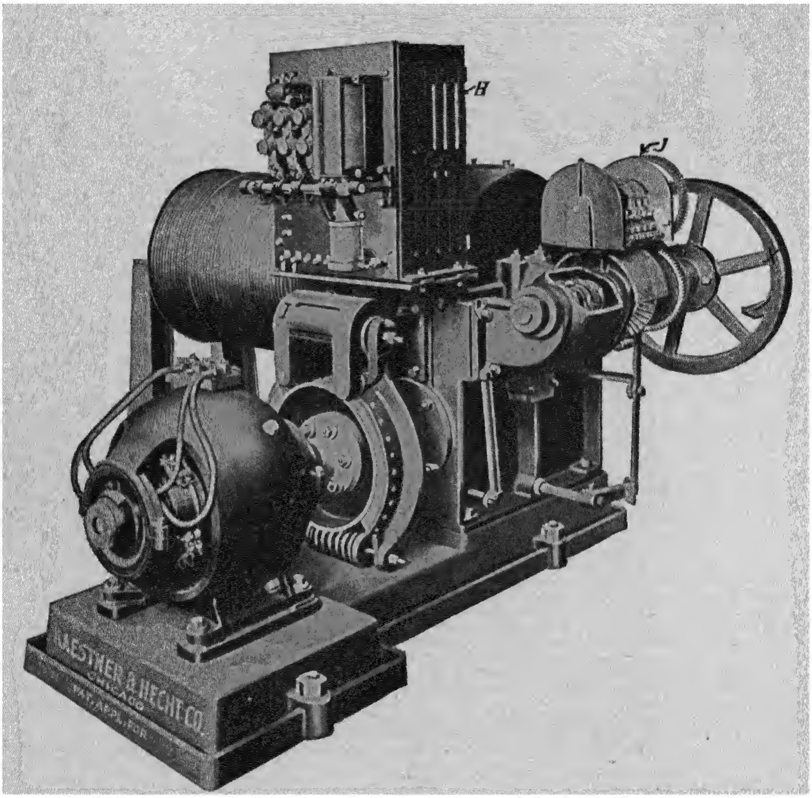



Fig. 126. Freight and Passenger Elevator for Direct Current Operated by Lever or Hand Cable
*Courtesy of Kaestner and Hecht Company*

ance in the field circuit, thus giving increased speed when needed; in the case of the interpolar motor previously described, it cuts in or out the series windings for the purpose of controlling the power of the motor to start up under load; finally, by introducing resistance into the armature circuit, a dynamic brake power is generated for use in slowing down the elevator and when lowering very heavy loads.

The controller as usually arranged consists of banks of resistance mounted on one side of a marble or slate slab, Fig. 125, on the opposite side of which are arranged the various solenoids and switches by means of which the different operations above described are carried out. This slate or marble slab is held upright by a suitable iron frame and holes are drilled for the passage of various wires and for the bolts used in attaching the parts to the slab. Marble makes the

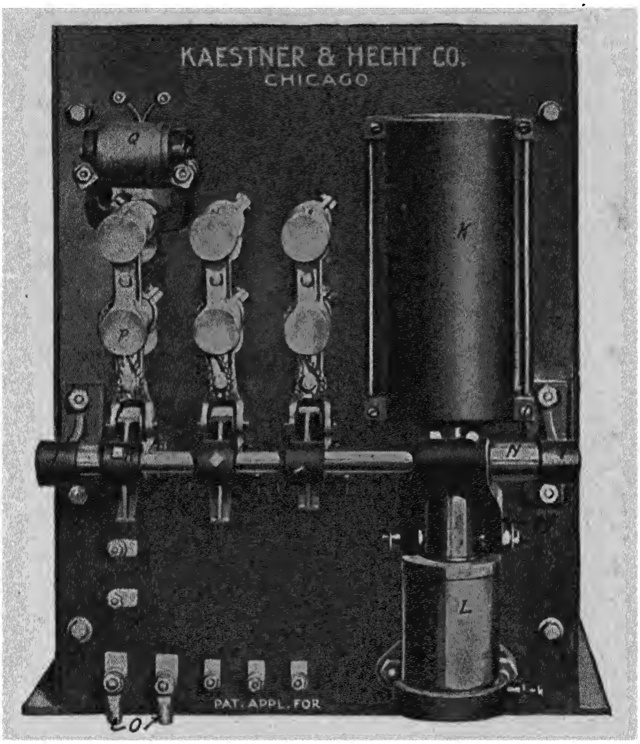



Fig. 127. Direct-Current Semi-Magnet Controller for Hand Cable and Lever Control Operation
*Courtesy of Kaestner and Hecht Company*

best switchboard material on account of its perfect insulating qualities, although slate is equally as good if free from streaks of metal, the presence of which will prove fatal to perfect insulation.

**Direct-Current Controller.** *Controller Proper.* A good example of the more simple direct-current type of controller is shown at *H* in Fig. 126, and in Fig. 127, which gives a front view on a larger scale, with the various solenoids and switches mounted in position. Re-

ferring to Fig. 127, *K* is the main solenoid, the energizing of which causes the plunger to rise and pull up the lever *M*, causing the shaft *N* to make a partial revolution; the cams *R* which are attached to the shaft *N* are, by its rotation, made to close the switches *P* consecutively (the cams being adjusted in proper position to do this), and, as each switch closes, it cuts out one bank of resistance and thus accelerates the car. The amount of time for accomplishing this is regulated by the dashpot *L*, a short tube filled with oil, containing a

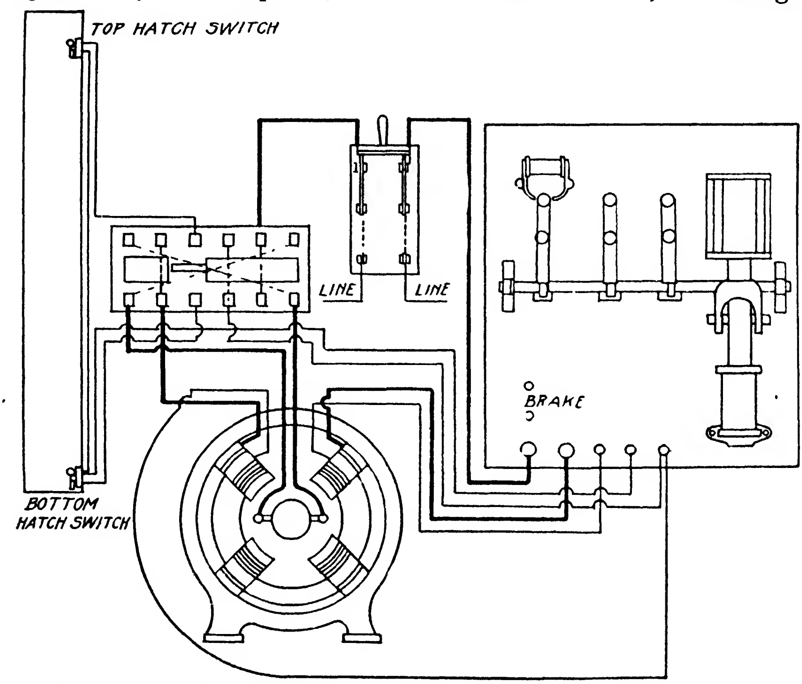



Fig. 128. Wiring Diagram for Semi-Automatic Controller for Compound-Wound Motor
*Courtesy of Kaestner and Hecht Company*

piston attached to the lower end of the solenoid plunger. By allowing the oil above the piston to escape through a graduated opening to the lower end of the cylinder, the upward motion of the plunger is retarded at will. The time required to close the three switches is from four to six seconds, depending on the nature of the work being performed by the engine. The moment the circuit is broken or opened for stopping, the plunger is quickly restored to its original position by means of a quick-opening valve working in one direction which allows the oil to return to its place above the piston. *Q* is a

magnet, the office of which is to blow out the arc caused by opening the last switch.

In order that the student may get a better idea of the circuits necessary to allow the controller to perform its functions, a few typical wiring diagrams will be introduced. Fig. 128 is a diagram of a semi-automatic controller for a compound-wound motor.

*Starting and Stopping Switch.* The opening and closing of the circuit for starting and stopping the motor are accomplished by means of the starting switch *J*, Fig. 126. This switch, which is of the rotating make-and-break type, is connected to the operating sheave by means of a pair of spur gears, and when the switch is made to revolve by means of the operating cable (which is passed around and attached to it), this closes the circuit in the direction it is desired that the elevator shall run. This controller performs but two functions: (1) cutting out the resistance in order to increase the current and bring about the proper speed; and (2) releasing the brake by energizing the solenoid *I*, Fig. 126, which pulls together the upper ends of the brake shoes, neutralizing the effect of the powerful spring connecting their lower ends, Fig. 129. Hence, the brake is applied at all times except when the solenoid *I* is energized. The wisdom of this arrangement is seen in case of accident or failure of the electric current, for the brake is then automatically set.

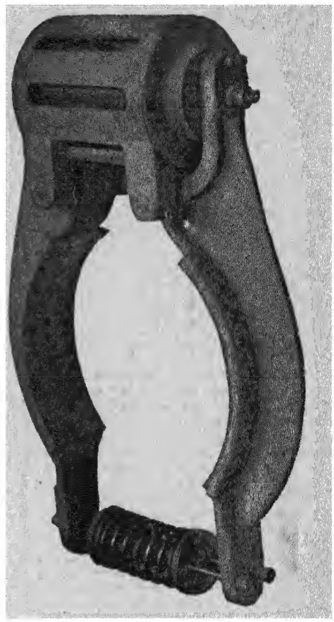


Fig. 129. Brake Shoe Operated by Solenoid

**Alternating-Current Controller.** *Mechanically-Operated Type.* When an alternating current is used for operating the elevator, different types of motor and controller are necessary. Only two- and three-phase motors have been found applicable to this problem, as no successful reversing alternating-current motor of the single-phase type has yet been made. As to the controller, the use of the solenoid on alternating-current circuits is not in general favor owing to the expense of production and its noisiness in operation. In these cases,

therefore, the makers have resorted to mechanical means for operating the brake and switches on the controller, while the cheaper and slower types are operated by hand cable or lever device at the car.

An electric engine of this type, Fig. 130, corresponds in every respect to that shown in Fig. 126, except that the brake *1* and con-

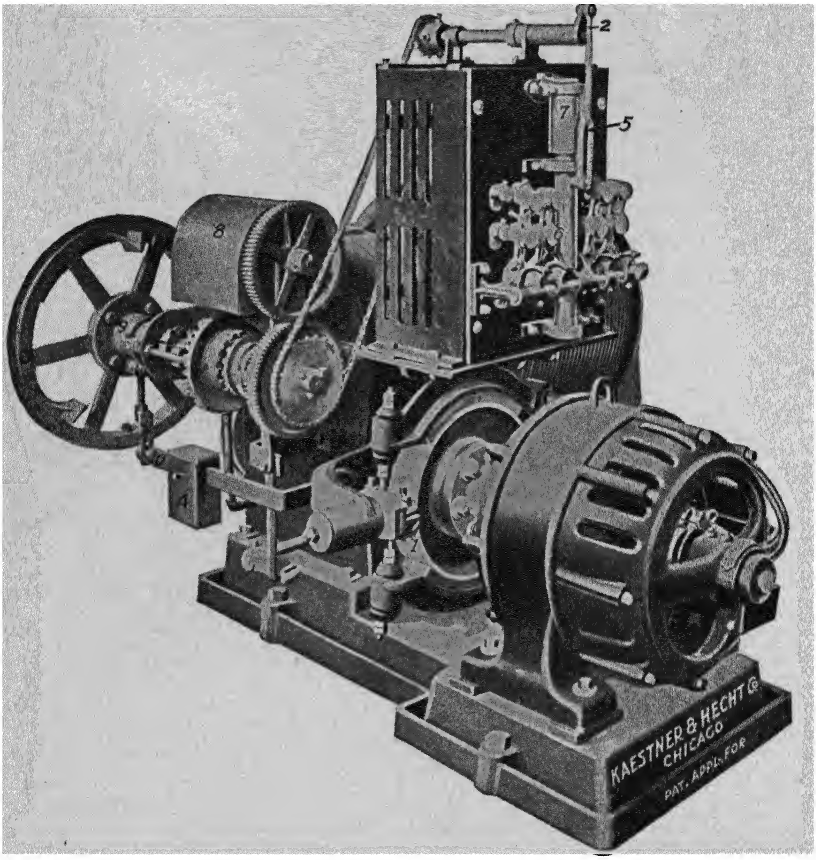



Fig. 130. Freight and Passenger Machine for Alternating Current
*Courtesy of Kaestner and Hecht Company*

troller *2*, also shown in Fig. 131, are mechanically operated and the motor is of the alternating-current type.

To operate the controller, one-quarter revolution of the operating sheave *3* causes the crank *2* to make one-half revolution, the ratio between the sprocket wheel on the end of operating shaft and that on shaft at top of controller being two to one. When the crank

*2* makes one-half revolution in either direction, the pitman *5* goes with it, leaving the plunger *6* free to descend by gravity. The motion of this plunger is retarded by the action of the dashpot *7*, in precisely the same manner as that for Fig. 127. The action of this plunger in dropping is practically the same as in the electrically-operated type, causing a shaft to revolve partially and, through the medium of cams, to close switches and thus cut out resistance from the motor. The making, breaking, and reversing of the circuit is done by the switch *8*. When the operating sheave is revolved in either direction to start the elevator, a cam at *9* lifts the brake lever *10*, thus

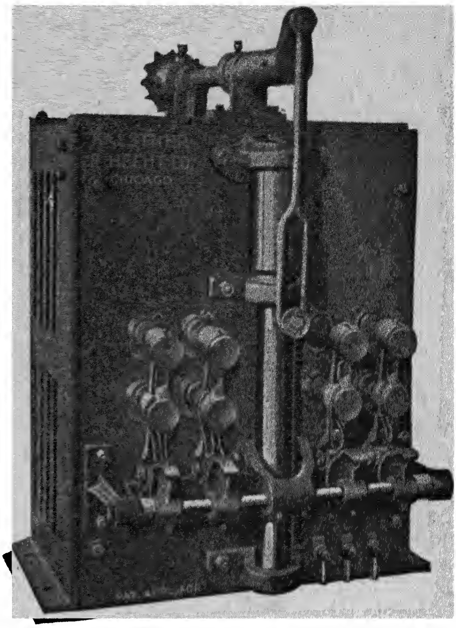


Fig. 131. Alternating-Current Mechanical Controller

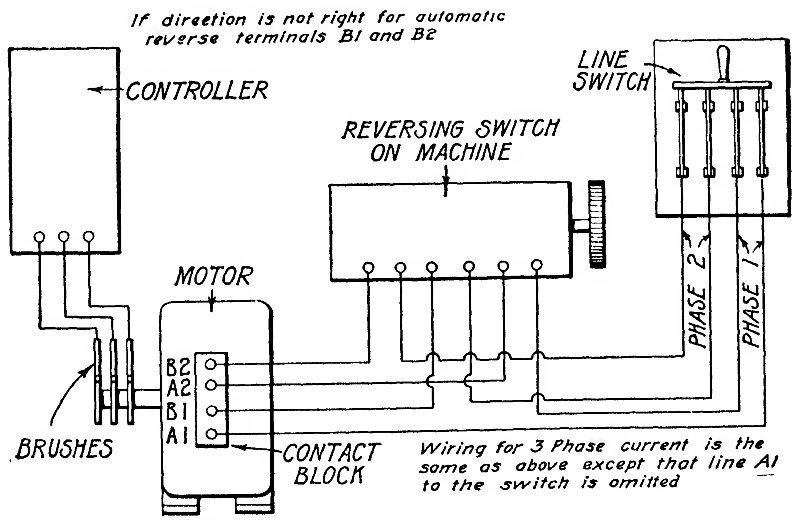



Fig. 132. Wiring Diagram for Two-Phase or Three-Phase Elevator Equipment with Mechanical Control

releasing the brake; and when the operating sheave *3* is returned to its original position, it drops the brake lever, and the weight *4* applies the brake by gravity.

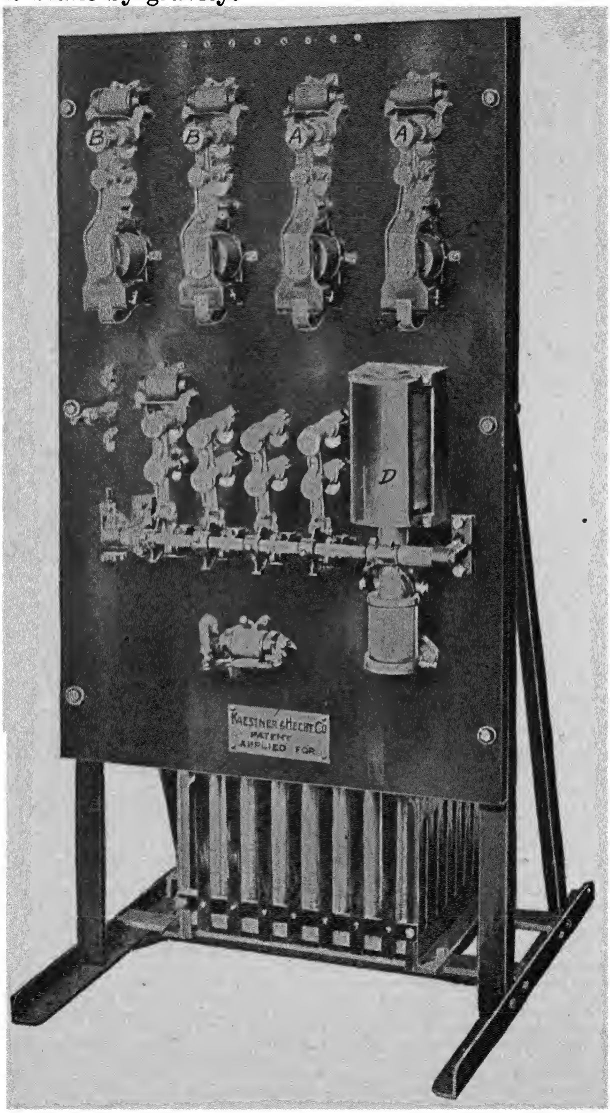



**Fig. 133. Single-Speed Full Magnet Controller**
***Courtesy of Kaestner and Hecht Company***

The wiring diagram, Fig. 132, shows the circuits for a two- or three-phase motor with mechanical control.

*Magnet Control.* When the motion of the car is governed by electrical means, the switches on the controller which open and close the circuit to the motor, are operated by means of electromagnets; hence the term "full magnet control". Fig. 133 is a good example of this kind of control where the mechanical operation remains un-

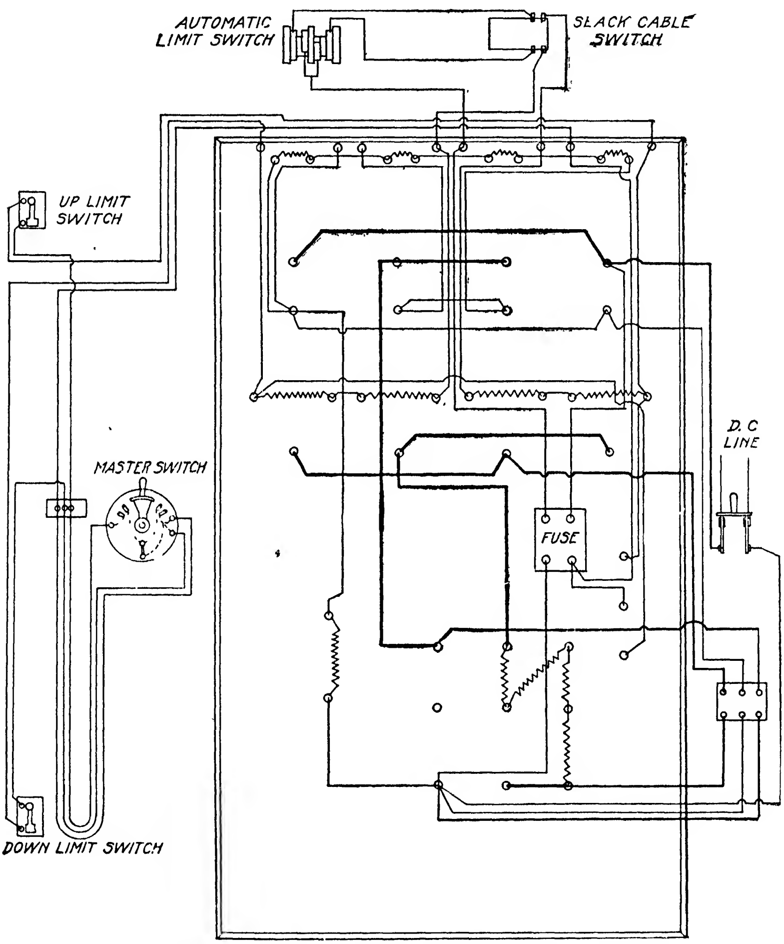



Fig. 134. Wiring Diagram of Single-Speed Full Magnet Controller

changed. *AA* are the switches used in closing the circuit for the *up* movement of the elevator, *BB* perform the same office for the opposite direction, and *D* is the solenoid for cutting out the several banks of resistance as the motion of the motor accelerates. The line

switches are closed by means of the electromagnets $C$, the springs $E$ automatically opening the switches when the current in $C$ is cut off. The wiring diagram for a single-speed full magnet controller is given in Fig. 134. Fig. 135 shows the method used by another manufacturer to accomplish the same purpose. Where a variable speed is desired, a controller, Fig. 136, similar to Fig. 133, is used; the only difference is in the use of the extra solenoid, shown at the lower left-hand corner of the board, and the accompanying switches for cutting resistance into the motor fields for accelerating

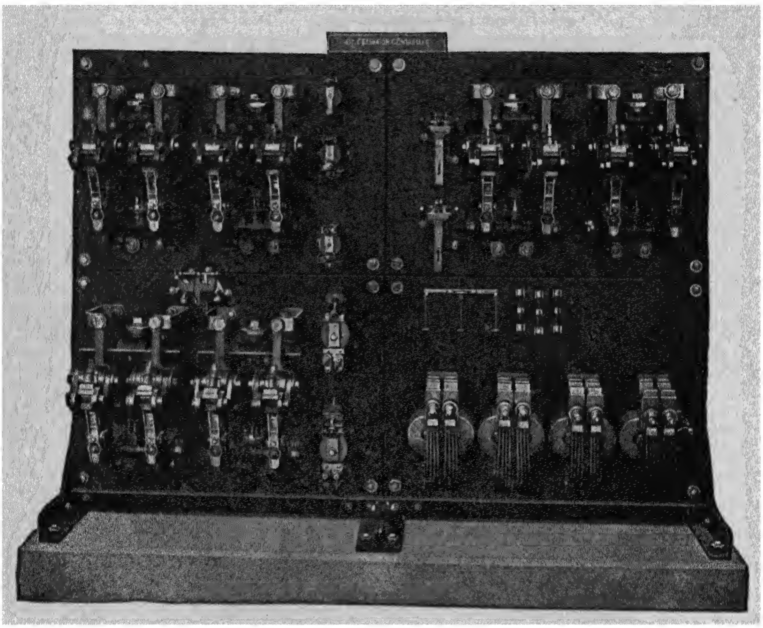


Fig. 135. Full Automatic Skip Hoist Controller for Alternating Current
*Courtesy of Otis Elevator Company*

the speed. This solenoid is energized by the movements of the car switch and is entirely under the control of the operator. The wiring diagram for a two-speed full magnet controller is given in Fig. 137.

Some elevator makers use a similar solenoid and switches on freight elevators, connecting the solenoid with the armature of the motor through special resistances so arranged that with light loads the solenoid is not energized, but under heavy loads the e. m. f. in the armature becomes great enough to move the plunger in the solenoid, which then operates the switches. In such cases, however, the

action of the solenoid is different, as the operation of the switches cuts out resistance from the fields, thus causing the motor to run at a

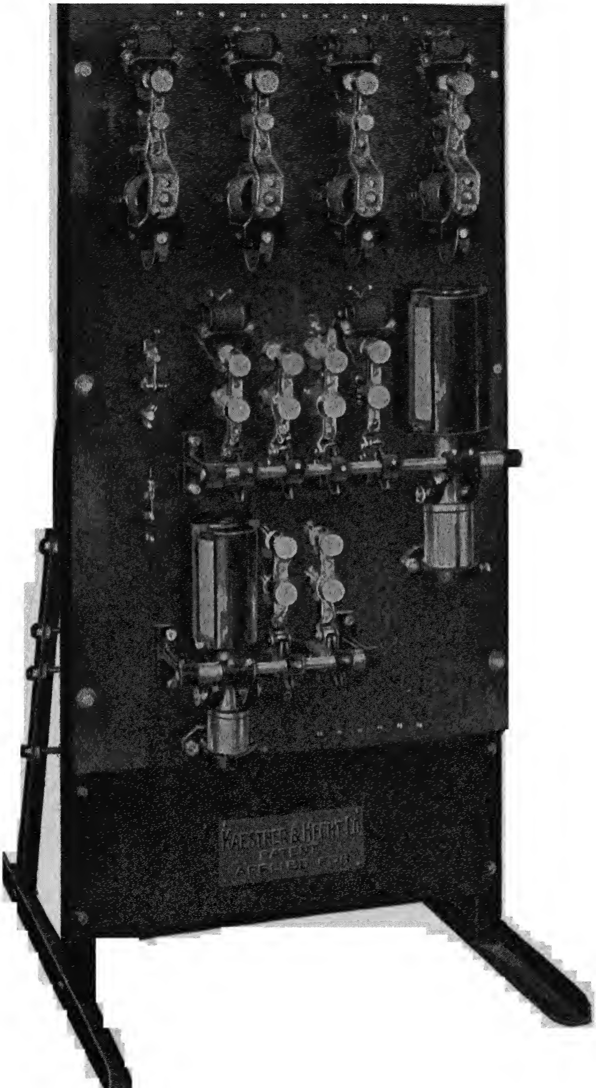


Fig. 136. Variable-Speed Full Magnet Controller
*Courtesy of Kaestner and Hecht Company*

slower speed—the field resistance is always in, except when the solenoid is energized. The effect of this arrangement is to produce

a fairly high speed with comparatively light loads; while with a heavily loaded car, the speed is automatically lowered by cutting out the field resistance.

Fig. 138 is similar to Fig. 136, except for two sets of switches for controlling field resistance, as well as a separate switch (the center

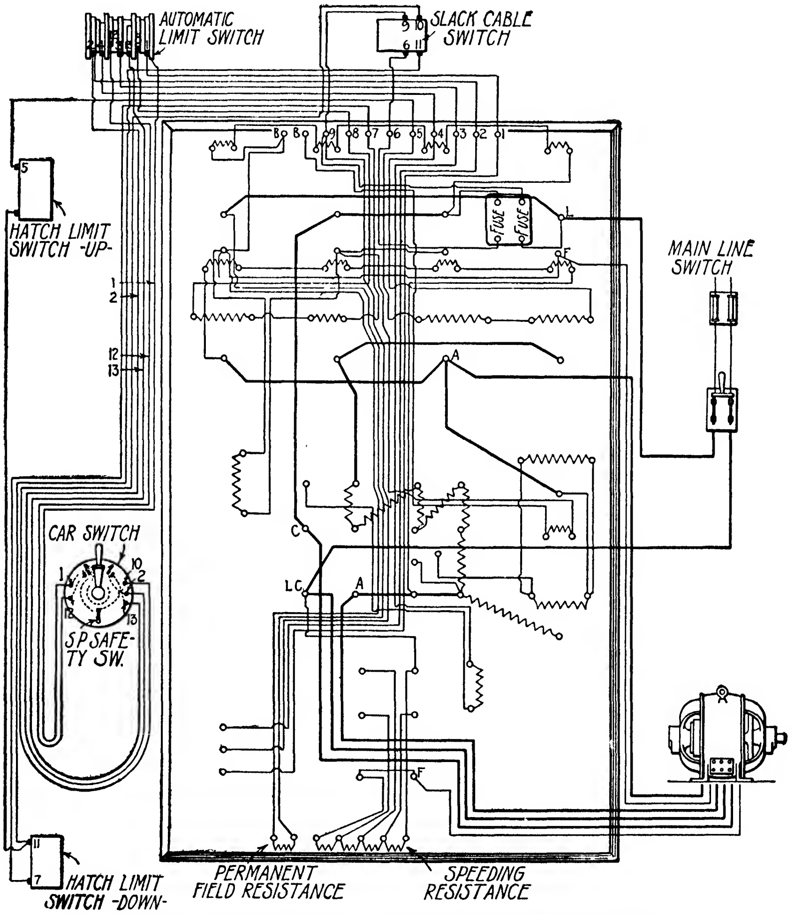



Fig. 137. Wiring Diagram of Two-Speed Full Magnet Controller

one, top row) for controlling the brake. This controller gives three speeds and is designed for use with motors equipped with interpole fields and heavy series winding, the action of which was described on page 169. Fig. 139 shows the wiring diagram for a three-speed full

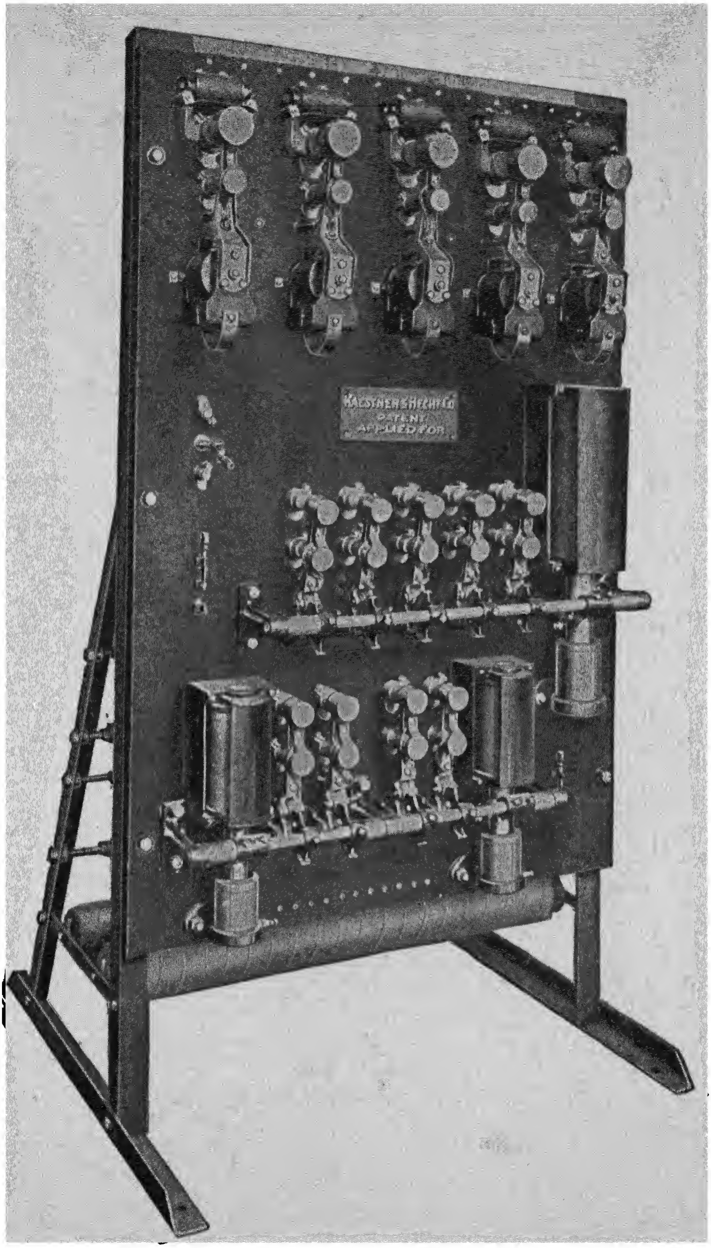



Fig. 138. Three-Speed Full Magnet Controller for High-Speed Passenger Elevator
*Courtesy of Kaestner and Hecht Company*

magnet controller with interlocking magnet and an extra line switch for 500 volts direct current.

*Push-Button Control.* MANIPULATION. Elevators using push-

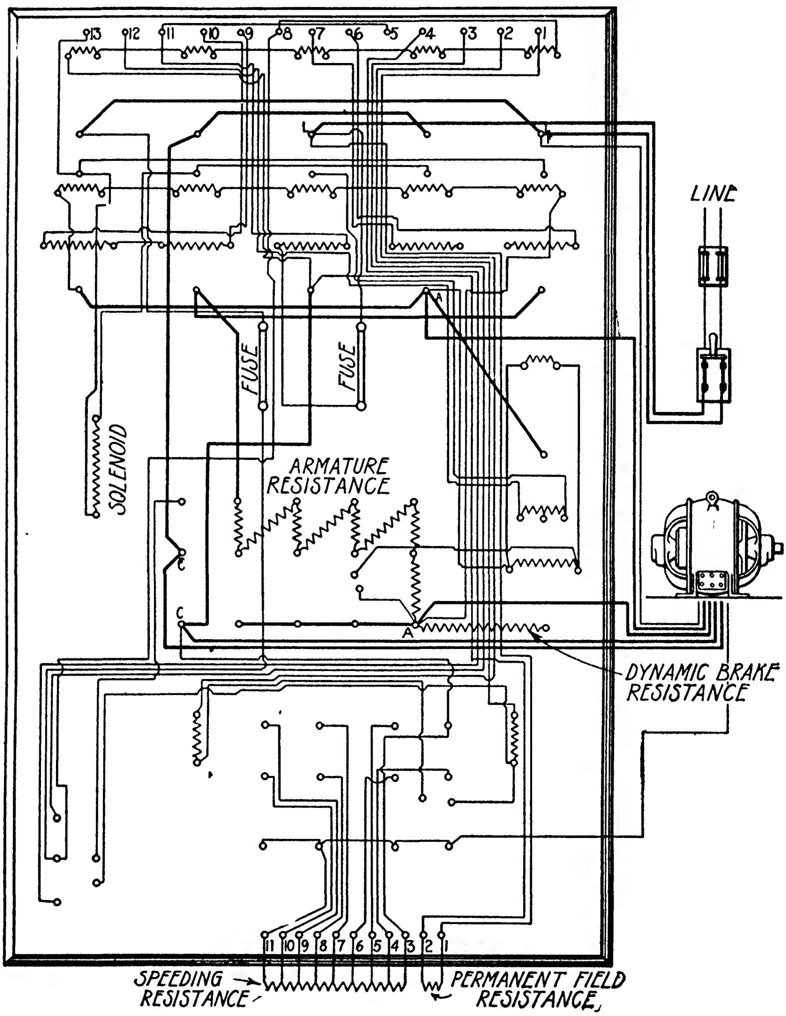



Fig. 139. Wiring Diagram of Three-Speed Full Magnet Controller with Interlocking Magnet and Extra Line Switch
*Courtesy of Kaestner and Hecht Company*

button controls are largely used in hospitals, apartment houses, and private dwellings. They are necessarily one-speed elevators and are intended to be operated by the passengers themselves. In Fig.

140 is shown a controller in its usual position near the engine; Figs. 141 and 142 show wiring diagrams for a.c. and d.c. current.

The manipulation of the automatic push-button elevator may be clearly understood from Fig. 143. The person desiring to use the elevator presses a push button near the elevator door (marked "Call Button"). The elevator at once comes to the floor at which the

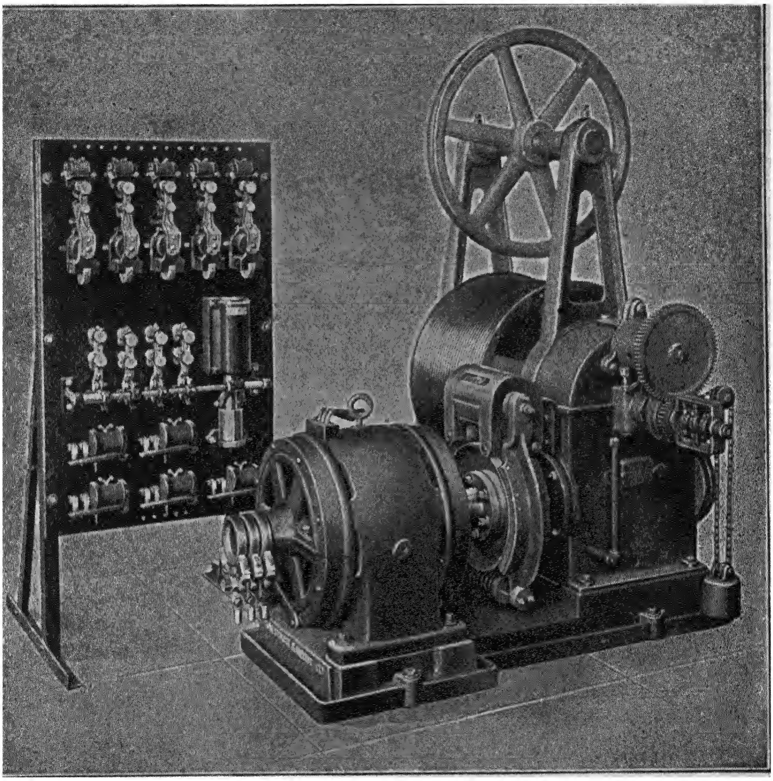


Fig. 140. Automatic Push-Button Controller and Machine
*Courtesy of Kaestner and Hecht Company*

operator is waiting and stops there. The door may now be opened, and after stepping inside the cab and closing the door, the operator selects from a bank of buttons inside the cab the one marked with the number of the desired floor and, upon pressing this button, the elevator moves to the floor designated and stops there. These push buttons are arranged one above another in a metal box, Fig. 144, and

are connected by means of a flexible cable with the necessary wires which run in the proper conduits to the hatch switches and to the controller.

Safety Doors. The doors to the shaft are so arranged that opening one of them breaks the circuit; consequently, if upon leaving the elevator at any floor the user fails to close the door after him, the

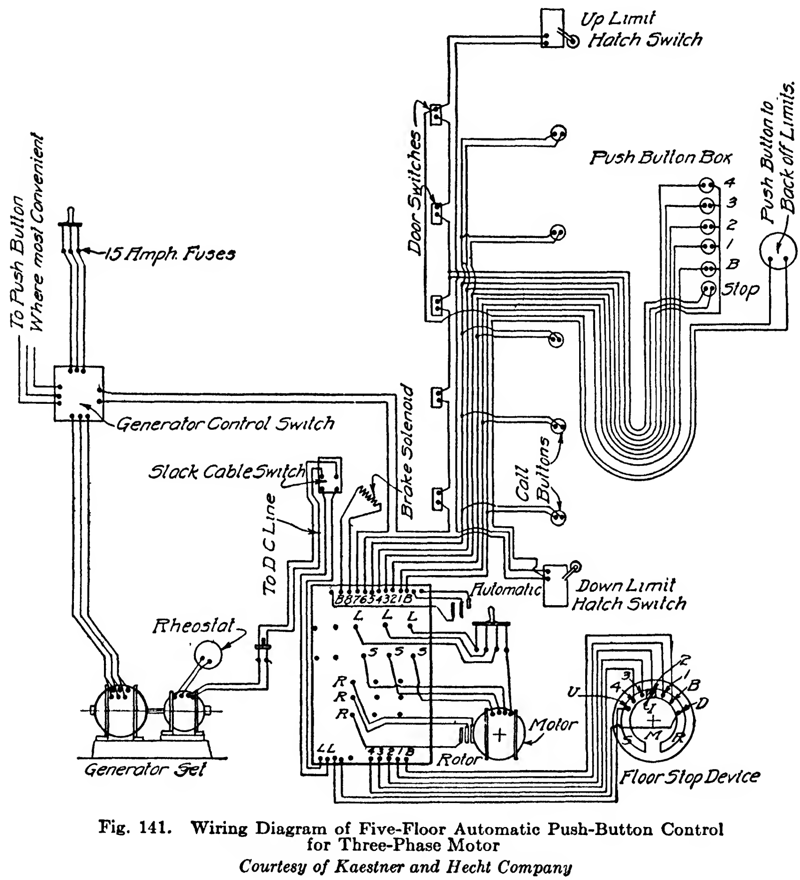



Fig. 141. Wiring Diagram of Five-Floor Automatic Push-Button Control for Three-Phase Motor
*Courtesy of Kaestner and Hecht Company*

elevator is out of service for the time being and cannot be started again until that door is closed. To prevent opening the elevator door when the car is not at the landing, a lock is provided which is released only when the car itself reaches the landing; this is accomplished by having a small cam on the car, which closes a switch, thus actuating a small electromagnet and unlocking the door. The

circuits for this arrangement are shown in Fig. 143 and are given with sufficient clearness to be almost self-explanatory. The floor

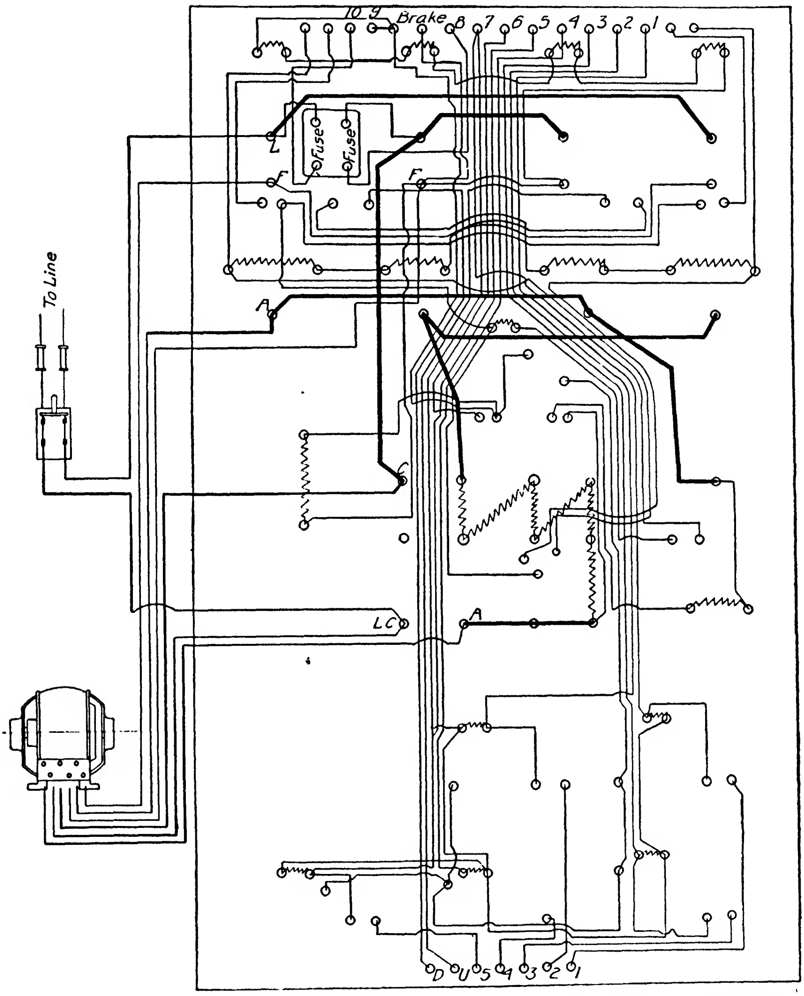



Fig. 142. Wiring Diagram of Five-Floor Automatic Push-Button Controller for D. C. Motor
*Courtesy of Kaestner and Hecht Company*

call buttons are on the right and the circuits leading to the car controller are clearly shown.

Up and Down Control. The up and down control of the push-button elevator, the wiring diagram of which is shown in Fig.

141, is accomplished by means of a floor-stop device as shown in the diagram. This device consists of an insulated disk *M* on which are mounted two brass annular segments *R* and *S* insulated from each other at *I* by a segment of insulating material. In front of the disk is a fixed frame containing brushes *D*, *B*, *1*, *2*, *3*, *4*, and *U*. These brushes are insulated from the frame and bear upon the annular segments of the disk as shown. The brush *D* is connected to the solenoid on the control panel operating the switches for a "down" direction of motion of the cage, while the brush *U* is connected to the solenoid operating the switches for an "up" direction of motion of the cage. The segment *R* may hence be called the "down" segment and the segment *S* the "up" segment in that they connect, respectively, with the switches for those particular directions of running.

The brushes *B*, *1*, *2*, *3*, and *4* correspond to floors and each is connected to a small electromagnetic switch shown at the bottom of the panel in Fig. 140, one electromagnetic switch being provided for each brush.

Fig. 141 is drawn on the assumption that the car is at rest at the second floor, in which case brush *2* is on the insulated segment *I*. Assume now that the fourth-floor call button or the fourth-floor button in the car is pressed. The momentary pressing of the button will operate the corresponding electromagnetic switch for that floor at the bottom of the panel. The switch will complete the circuit of the solenoids operating the main switches for an upward movement of the cage, through the brush *4*, the "up" segment *S*, and the brush *U*. The car will hence move upwards. In so doing the disk *M* will be caused to revolve in a counterclockwise direction, as it is connected to the cable drum or a cable sheave by means of a chain and sprocket wheels which give the disk a motion proportional to the motion of the cage and a direction of motion dependent upon the motion of the cage.

Hence the rotation of the disk *M* will cause the insulated segment *I* to move to the left. It will then pass under the brush *3*, and after its passage the brush *3* will bear on the "down" segment *R*, but the car will continue to move upwards, as the "up" switches were closed through brush *4*. When brush *4* bears on the segment *I*, it, of necessity, ceases to bear on the "up" segment *S*, and hence

the circuit to the "up" switch solenoid and the brake-release solenoid is broken and the car stops. Now the disk is so geared to the cable

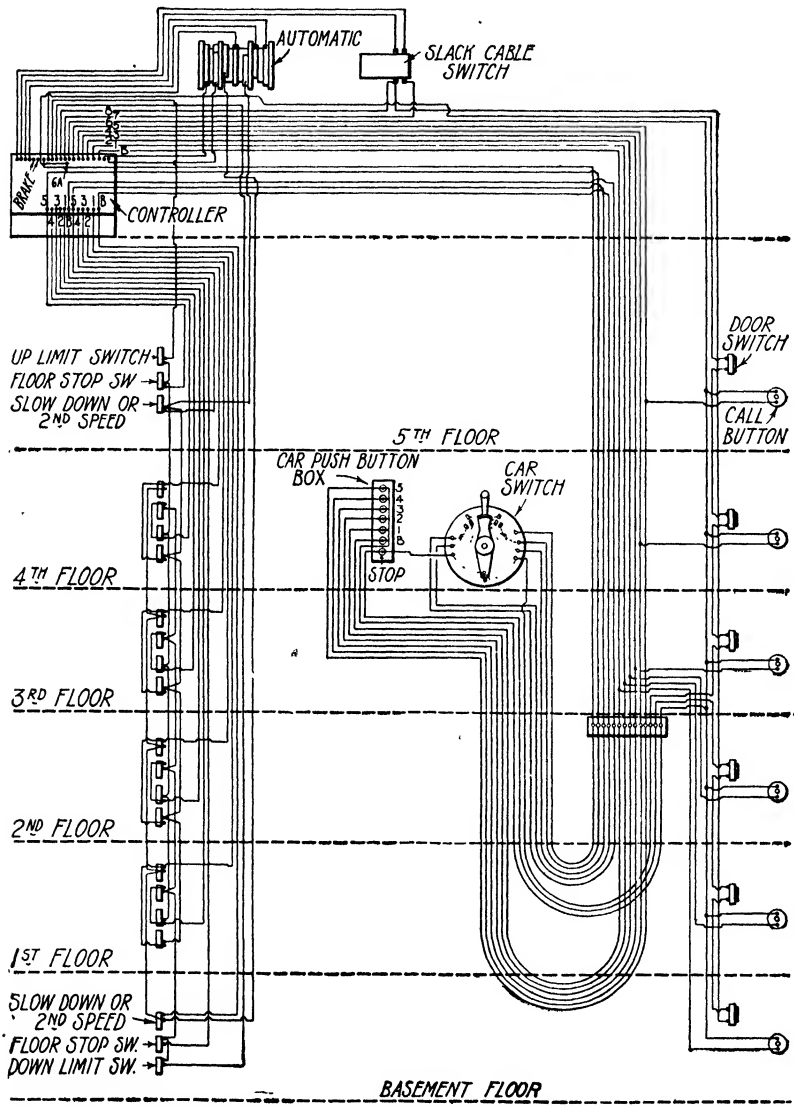



Fig. 143. Hatch Wiring Diagram of Six-Floor Combination Two-Speed Car Switch and Push-Button Control Elevator

*Courtesy of Kaestner and Hecht Company*

drum or to a cable sheave by sprocket wheels and a chain that when a brush is on the segment *I*, the car is at the landing corre-

sponding to that brush. Hence in the case under consideration the car will stop at the fourth floor. It is seen that by this arrangement the brushes *B*, *1*, *2*, *3*, and *4* are made to pass from one segment to the other as the car passes a floor, hence placing each of the brushes on the segments that will give the car the correct direction of motion for traveling to the floor desired by the passenger, when the correct push button is pressed.

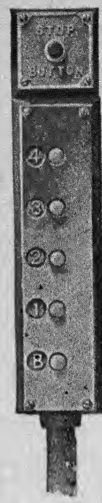


Fig. 144. Warner Cab Push Buttons

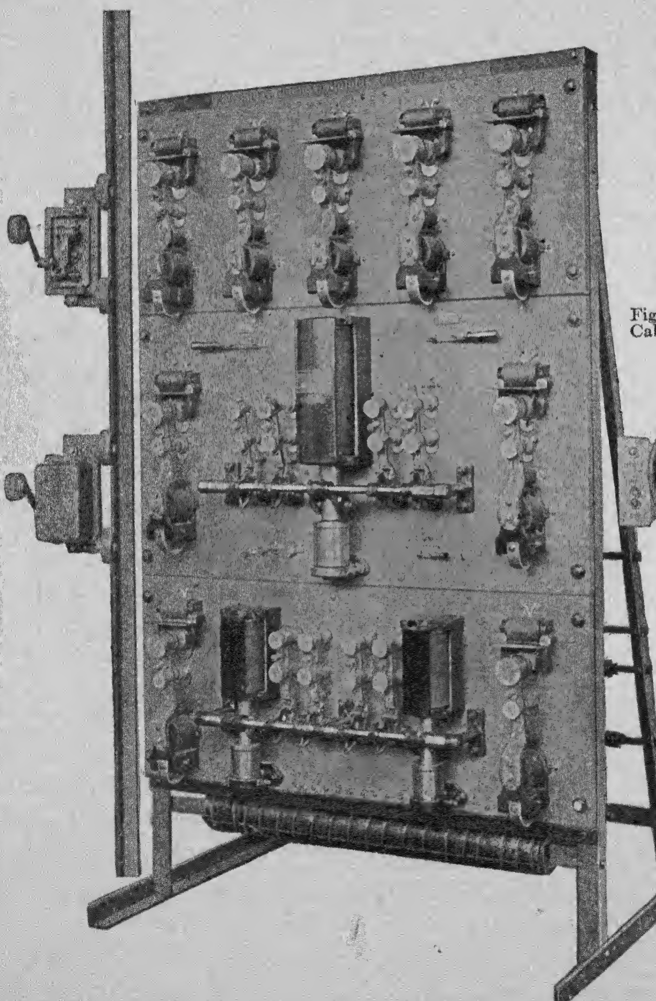


Fig. 145. Variable-Speed Full Magnet Elevator Controller
*Courtesy of Kaestner and Hecht Company*

The preceding description makes clear the method of push-button control when an alternating-current motor is used. In those cases where the direct current is used exclusively for operating the elevator, extensive modifications of the circuits must be made, as will be seen from the diagram shown in Fig. 142. However, the method of control is essentially the same as far as the manipulation in the car and at the various floors is concerned.

Referring to Fig. 140, it will be seen that there are five switches arranged at the top of the controller board; the two outer ones on either side are the line switches, while the center one is for closing

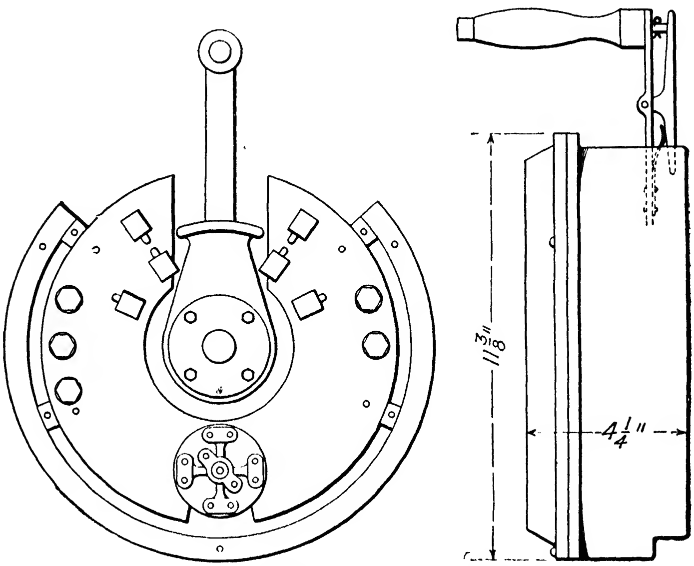


Fig. 146. Front and Side Elevations of Car Master Switch

the circuit to the brake. The four smaller switches and solenoid just below are as usual for controlling the acceleration of the motor. The five small spools at the bottom of the board are electromagnets for closing the line and brake switches and are operated by the push buttons and disk switches. As one of these small electromagnets is required for each floor, the machine shown in the illustration is a five-floor controller.

The bank of push buttons in the cab comprises one button for each floor, appropriately numbered, and one button marked "STOP", Fig. 144, to be used in case the operator makes a mistake. If he

selects the wrong floor, he simply pushes the "STOP" button and makes his selection anew.

Motor. The motor, Fig. 140, is an alternating-current three-phase motor. It might be mentioned here that any single-speed elevator may be operated on alternating current with magnetic control by using a small motor-generator set comprising an alternating-current motor and a direct-current generator to generate sufficient direct current to operate the electromagnets and solenoids for the control. (See the circuits for this arrangement at the lower left-hand corner of Fig. 141.) This small transformer set would run constantly and need not produce more than 10 to 20 amperes at 110 volts, the amount of current depending upon the capacity of the elevator. Most of the current would be used by the solenoid which releases the brake, the rest of the control never requiring more than 4 amperes.

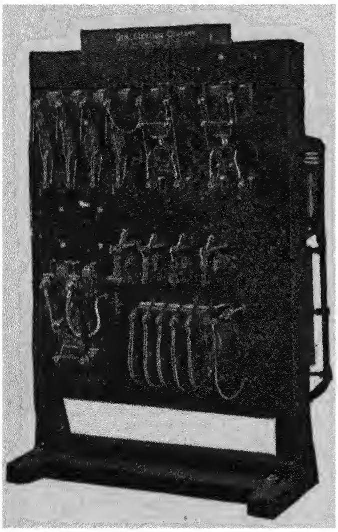


Fig. 147. Otis Direct-Current Elevator Controller

*Variable-Speed Controller.* The controller used for variable-speed elevators, running from 300 to 600 f.p.m., is shown in Fig. 145. Of the five switches located on the upper panel, the two outside ones on either side are the line switches, as before, while the center switch manipulates the field. In the center of the middle panel are the solenoid and the switches for cutting in and out the resistance in starting up on the first speed; the switch to the right in the same panel operates the dynamic brake; and that on the left operates an extra set of shunt fields used in connection with high-speed elevators for the purpose of giving still greater torque at starting, these shunt fields being auxiliary to the regular shunt fields of the motor. In the lower panel are located the switches and cut-outs of the two other speeds used in accelerating after starting.

To the right of this controller board is the operating switch which is mounted in the car. This is shown diagrammatically in Fig. 146. Moving the handle to the right produces motion of the car

in one direction; and to the left, the reverse. It is so constructed that should the operator remove his hand from the lever, it will automatically assume the stop position shown in the cut. Just below this lever near the lower part of the case is a button, Fig. 145, which operates the emergency switch for opening the circuit in case of accident to the lever or its connections. At the left side of the case and attached to it is a small white panel carrying the reset for the circuit breakers which are used in connection with this type of elevator. These circuit breakers, in case the elevator should be overloaded or the voltage on the line become either abnormally high or low, will of themselves automatically open the circuit, thereby protecting the motor from the danger of burning out.

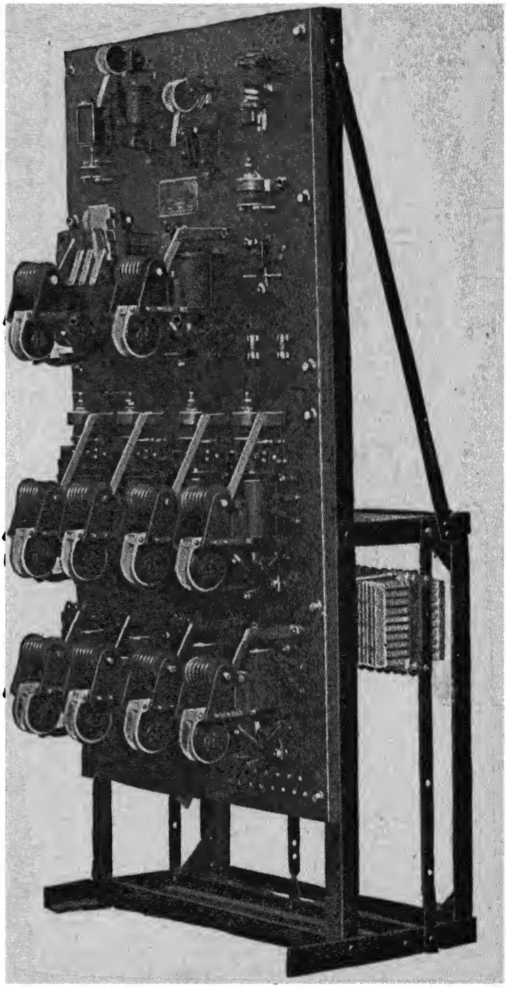


Fig. 148. Haughton Type of Controller
*Courtesy of Haughton Elevator and Machine Company*

At the extreme left of the illustration is a section of one of the steel guide rails in which the counterpoise weight travels. This has attached to it two hatch switches, already referred to as being used for the purpose of slowing down and stopping the car at each end of its run. A cam attached to the car presses back the lever shown at the side of the switch, thus producing the desired results. The upper switch is shown with the cover removed. These switches are clamped to the guide rail in such a way that they may be readily

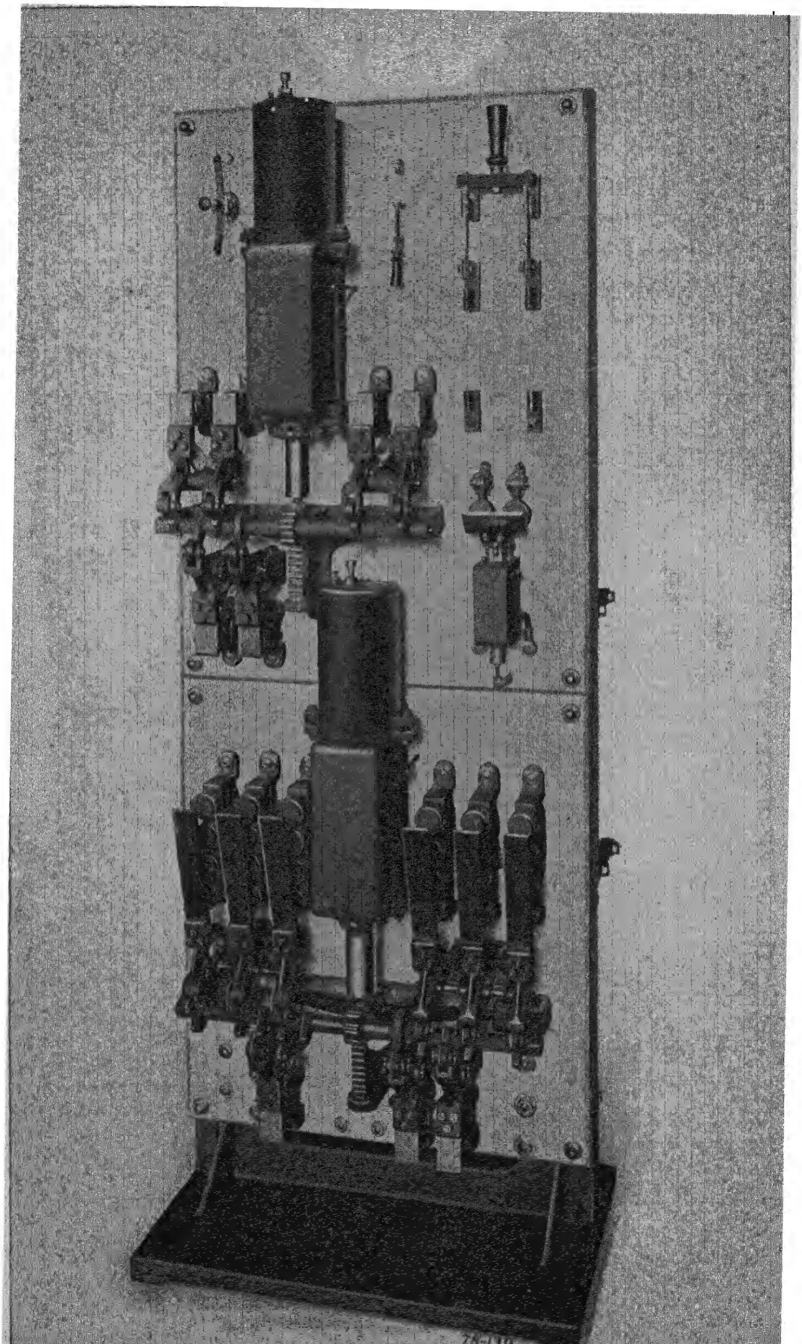


**Fig. 149. Gurney Type of Controller**
*Courtesy of Gurney Elevator Company*

adjusted for position with respect to the upper and lower landings, and in actual use are set several feet apart, instead of close together, as shown in the illustration.

A close observation of the panels of the two- and three-speed controller (Figs. 136 and 138) will show one or more single-pole knife switches. These are used for the purpose of throwing out of service the extra speeds, thereby converting the machine into a single-speed elevator for the time being; by this means an elevator designed to lift a moderate load at a high speed may be used for lifting a heavy load at about half speed. This arrangement, while useful for an occasional load, is not intended nor is it suitable for continuous service, being liable under such conditions to overheat the motor windings.

For the foregoing illustrations we have selected representative types of one of the best manufacturers. To illustrate differences in design, controllers of other standard makes are shown in Figs. 147 148, and 149.

## TRANSMISSION

The motor and controller having been discussed, only the transmission needs to be considered in order to complete what is usually known by elevator builders as the *electric engine.*

**Location of Engine.** The different forms of transmission vary somewhat with the location of the engine. This is, in some cases, placed on a foundation in the lower story, Fig. 150, or alongside the hatchway on any other floor of the building, but the most usual location is directly over the hatchway or shaft, Fig. 151. A penthouse is made especially for it above the roof, the engine resting on a floor supported by steel I-beams which, in turn, are carried by the walls of the building and the hatchway. The engine is securely bolted to these beams and slots are cut through the floor for the passage of the cables which connect the winding drum with the car and counterweights. The latter arrangement has the advantage of a more direct connection between the engine and the car and the elimination of the overhead sheaves which are necessary when the engine is located below and adjacent to the hatchway. The placing of the engine overhead does not always do away with the necessity for overhead sheaves for, where the hatchway is large or the necessary location of the engine is such that the diameter of the winding will

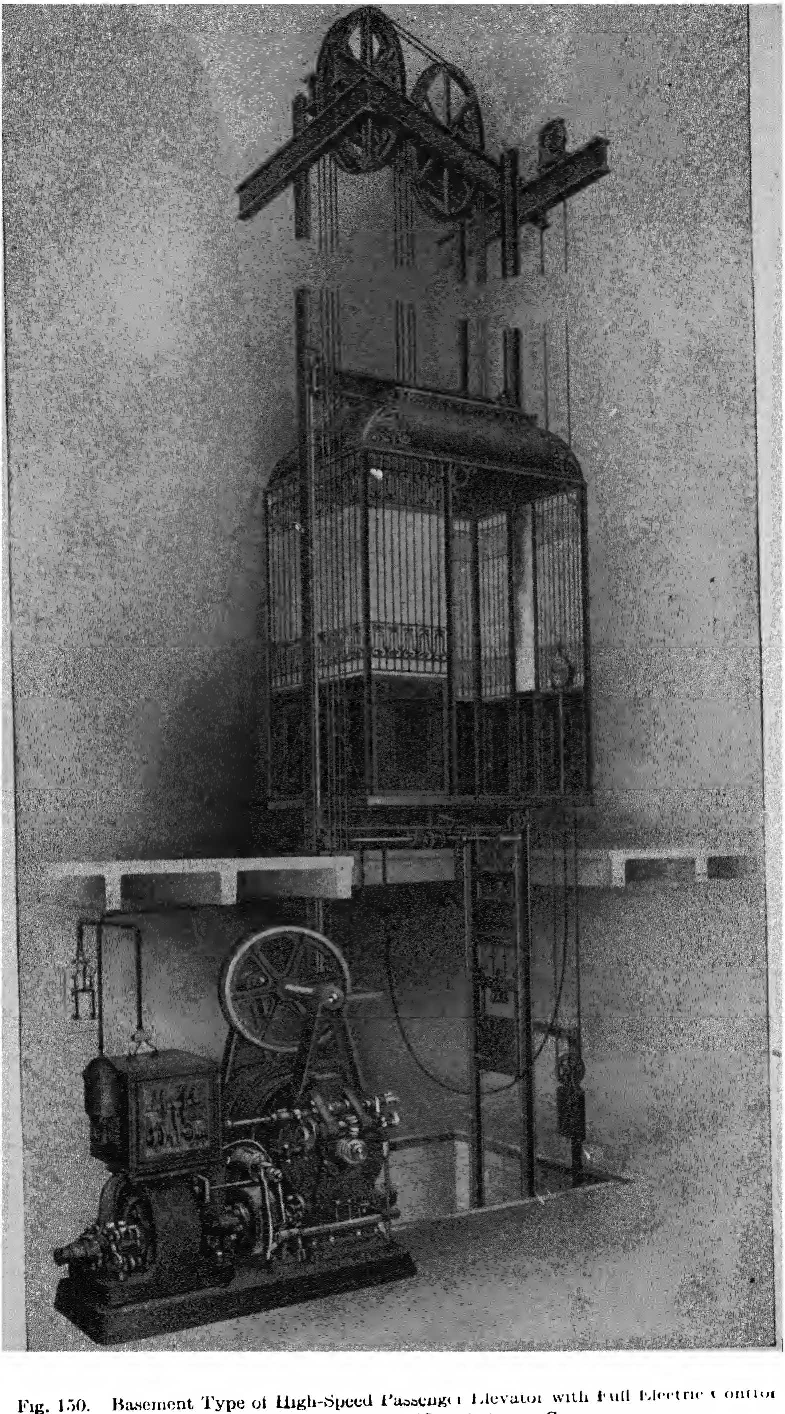


Fig. 150. Basement Type of High-Speed Passenger Elevator with Full Electric Control
*Courtesy of Warner Elevator Manufacturing Company*

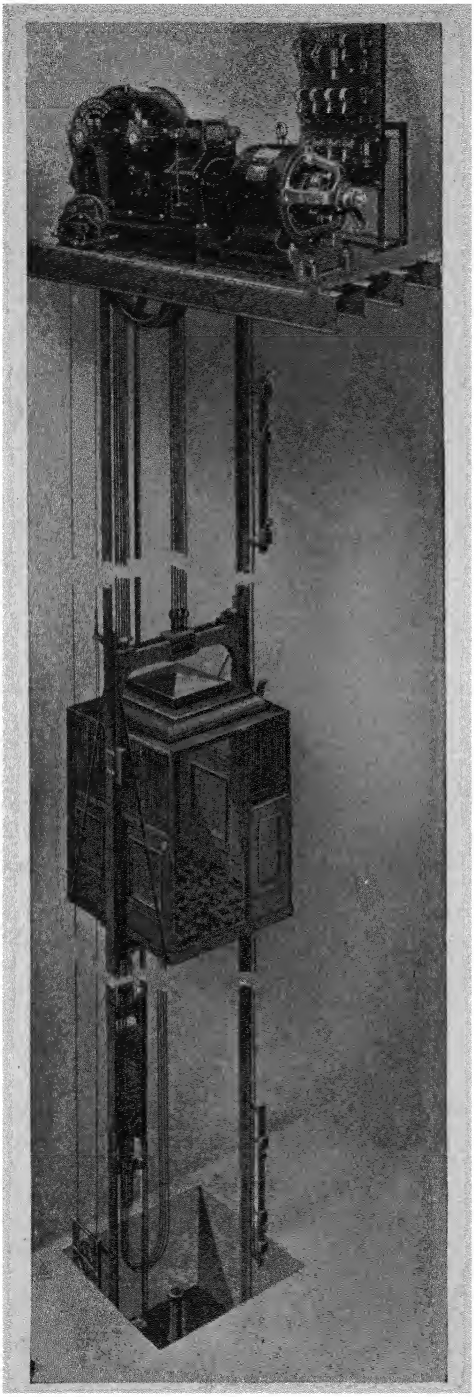


Fig. 151. Overhead Type of Passenger Elevator for Heavy Duty with Full Magnet Control
*Courtesy of Haughton Elevator and Machine Co.*

not span the distance from the center of hatch to center of weight slides, auxiliary sheaves become a necessity.

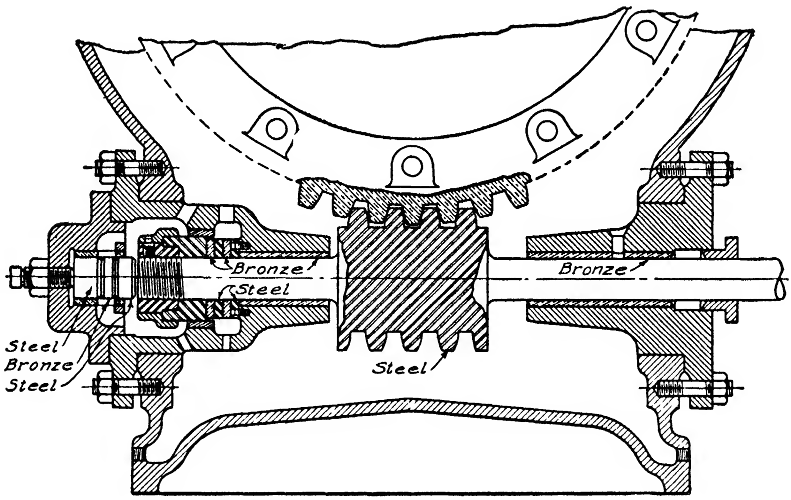



Fig. 152. Single Gear Worm and Wheel

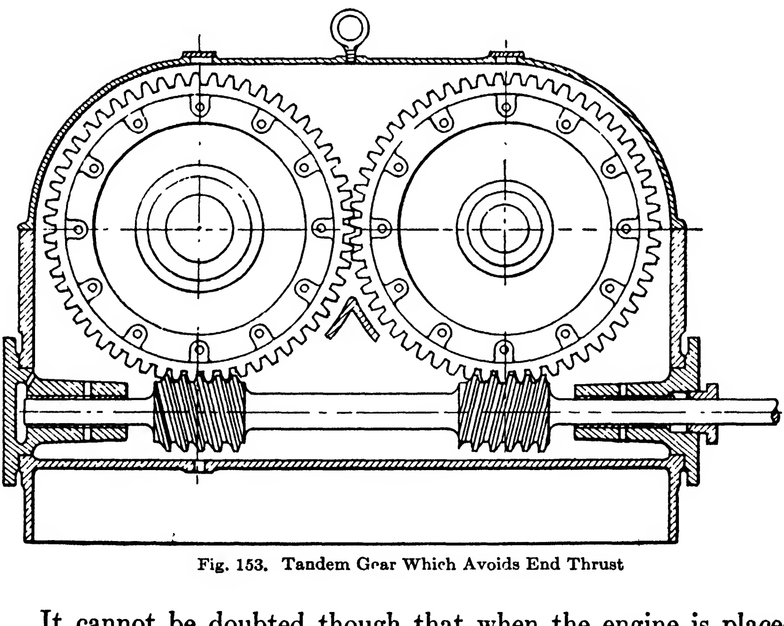


Fig. 153. Tandem Gear Which Avoids End Thrust

It cannot be doubted though that when the engine is placed overhead, a great deal of friction is dispensed with, and some room

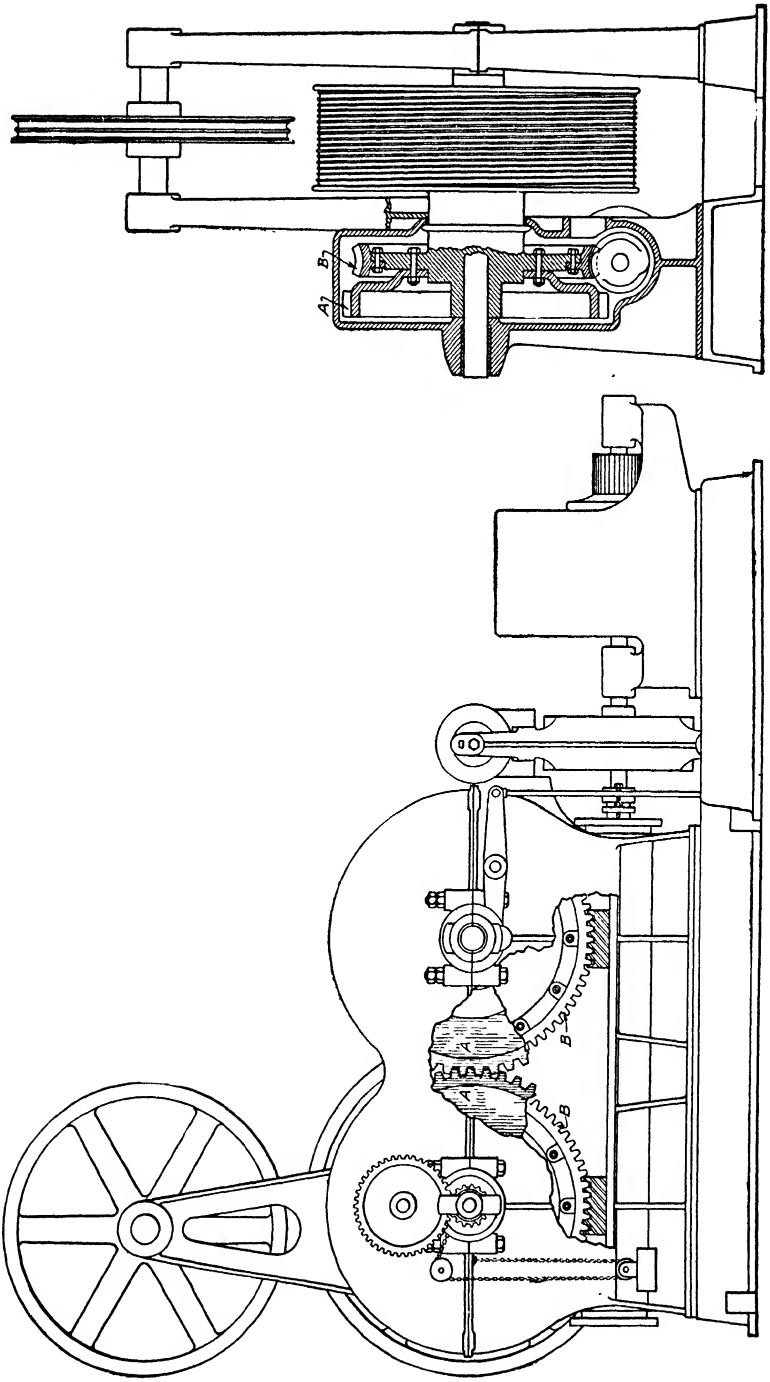



Fig. 151 Early Design of Spur Gears and Tandem Worm
*Courtesy of Kaestner and Hecht Company*

in the building saved for other uses. This arrangement calls for either brick walls surrounding the hatchway or girders at the roof sufficiently strong to carry the weight of the engine, car, counterpoise weights, and loads to be lifted, as well as the weight of the wall and roof of the pent house. The support must also be strong enough to resist the impact caused by the stopping and starting of the elevator.

**Worm and Wheel.** *Single Gear.* The power of the motor is transmitted to the winding drum and thence to the car or platform by means of a mechanism called a worm and wheel, Fig. 152 The shaft on which the worm is cut is direct-connected to the armature shaft of the motor by means of an insulated coupling (see Fig.

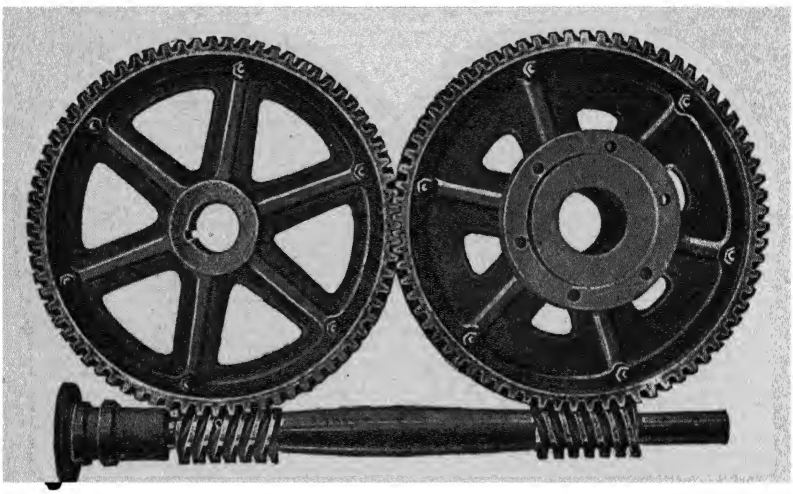


Fig. 153 Gurney Tandem Worm Gear Drive

126), one-half of which is shaped like a pulley of very heavy design and liberal width of face. This coupling serves as a brake pulley for use in stopping the machine and for holding it while being loaded and unloaded.

*Tandem Gear.* Another type of worm gear—called the tandem gear—is quite an old idea, having been patented over thirty years ago by a Boston firm. It consists of two worms or coarse pitch screws, Fig. 153, usually forged solid on one shaft; one screw being cut with a right-hand thread and the other with a left-hand thread. As originally designed by the inventors, the worm gears used were made with straight faces—the teeth being cut at the angle of the

thread of the worm—and the gears were set so that they meshed together. Later some makers used spur gears in addition to two worm wheels, Fig. 154, and bolted them together in pairs. The wheels were made with concave faces and teeth, as shown in the end elevation at the right, and the spur gears were enough larger in diameter than the worm gears to prevent the latter from touching each other when the spurs were in mesh.

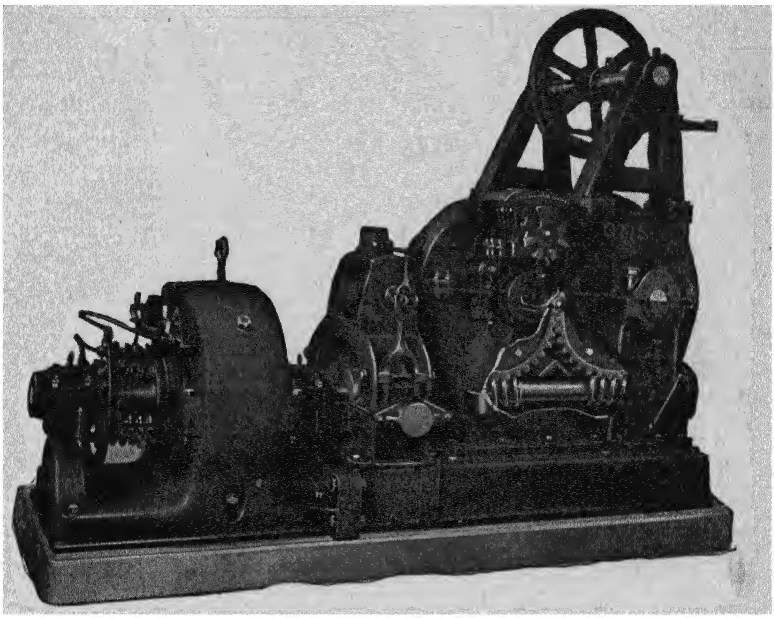


Fig. 156. Double-Screw Direct-Current Elevator Machine with Switch Control
*Courtesy of Otis Elevator Company*

Another and better way to secure these advantages is to use two spiral gears meshing together with the worms driving them, as shown in Fig. 155. Fig. 156 gives the general appearance of a tandem-geared engine.

The advantages claimed for the tandem-geared engine are double power and durability. The load, being divided between two worms and gears, produces only one-half the strain on the teeth of each, but it is a more expensive machine to build and occupies more space. However, for heavy loads the arrangement is certainly preferable to a single-geared machine having an enormous gear and

drum. Another advantage is that the gears used in the tandem machine are about the same size as those in general use, and therefore no special patterns or tools are required for the production of a machine of double capacity except the gear case and the bed, or base plate, of the engine. This tandem construction also eliminates the end thrust, which is discussed on page 204.

**Provisions for Speed and Load Variations.** *Change of Gear Pitch.* In all these machines it is customary to so design the gear cases that a choice of any of two or three pitches of gear and worm may be used, the range for the smaller sizes being from $\frac{7}{8}$-inch to $1\frac{1}{2}$-inch pitch, Fig. 157, with lifting capacity of 1500 to 3000 pounds, and from $1\frac{1}{4}$-inch to $1\frac{3}{4}$-inch for the heavier capacities. As the worm gear only moves the space of one tooth for each revolution of the worm shaft, the latter must revolve as many times as there are teeth in the gear for each revolution of the drum; so if the drum, in order to make the car travel at the required speed, had to make 15 r. p. m. and the worm gear had 50 teeth, the speed of the worm shaft and consequently that of the motor would have to be 750 r. p. m. A different speed of the drum may be brought about by a choice of any one of two or three pitches of worm gear. Of course, the same result could be accomplished by modifying the diameter of the drum, but this has some limitations in that the drum must not be so small in diameter as to have a bad effect on the cables, nor so large that the pressure between the teeth of the gear and threads of worm will cause excessive wear.

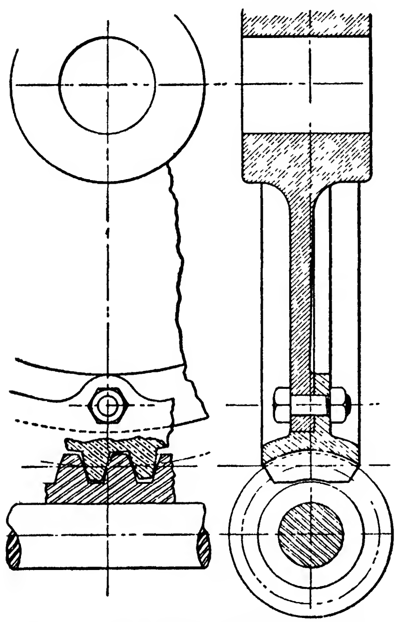


Fig. 157. Diagram Showing Arrangement for Changing Gear Pitches

*Double-Threaded Worm.* Where greater speed of car is desired, a double-threaded worm may be used, thus giving double speed to the drum. Therefore, in a machine built so as to allow a choice of any

one of three gears and worms, and, in addition, the double pitches just spoken of, there are six available speeds. These are capable of further variation either by the use of high-speed or low-speed motors or by varying the diameter of the hoisting drum, thus providing a possible variation of car speed of from 100 to 200 f. p. m.

*Reduction Gear.* When a car speed of from 50 to 60 feet is desired, it is customary to reduce the number of revolutions of the drum by means of a reduction gear, Fig. 158. The gear used in this machine is a spur of the "internal" type and, as the pinion usually

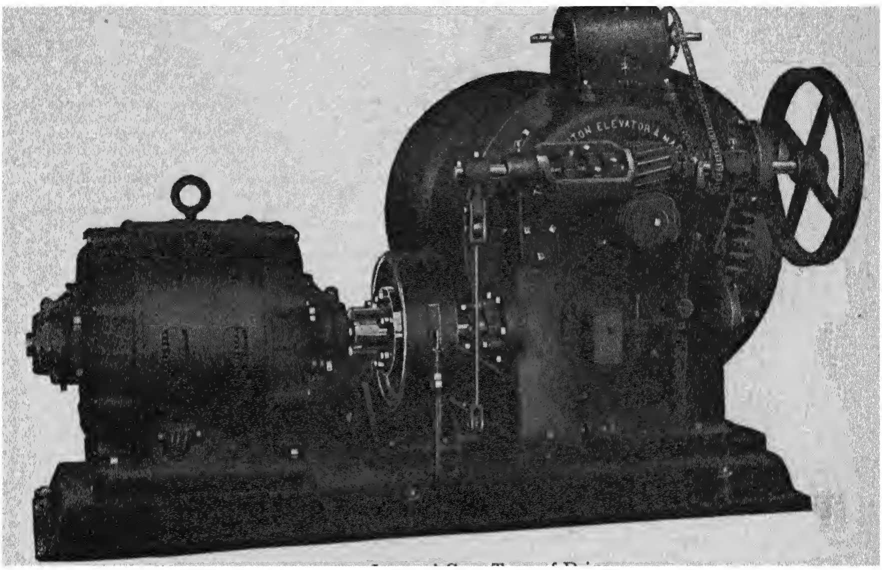


Fig. 158. Internal Gear Type of Drive
*Courtesy of Haughton Elevator and Machine Company*

has a ratio of about 4 to 1, a high speed motor may be used. Of course when the reduction gear is used, the pinion is placed on the same shaft as the worm gear and the drum runs in separate bearings. The principal reason for the use of an internal spur gear is that it does not change the direction of the drum's rotation, while a spur gear of the ordinary type would. It is always best that the drum in lifting the load should revolve so as to bring the thrust of the end of the worm shaft towards the back end of the gear case.

**End Thrust of Worm Shaft.** In order that the reader may have a clear idea of the action of the end thrust of the worm shaft in a

direct-connected single-gear machine, the details of the back bearing are given in Fig. 159. An examination of Fig. 152 shows that this end thrust is a very important thing, as upon the resistance of the back bearing depends the lifting power of the machine. The bearing shown is the most common form of thrust bearing and is so arranged that the end thrust is capable of adjustment in both directions. For the back thrust, a steel toe or thrust plug with a taper shank fits into a hole bored in the end of the worm shaft at *B*, Fig. 159. This is held in place by another steel plug *C*, with a hard bronze disk

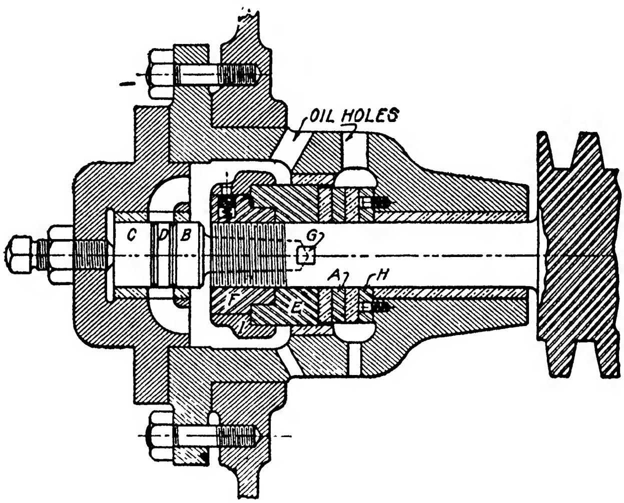



Fig. 159. Section of Worm Shaft Thrust Block

*D* between them. This steel plug *C* is adjustable for wear by means of a set screw, and is kept from revolving by a pin, not shown. The bronze disk, however, is free to revolve with *B* and bears the brunt of the wear.

The forward thrust due to the action of the counterpoise weights is taken on the alternate steel and bronze rings *A*, which are also free to revolve; the steel collar *E* is kept from revolving by the pin *G*, but has a longitudinal slot in it which fits over *G* and allows some end motion. The nut *F* is screwed on the end of the worm shaft and, by tightening up this nut when necessary, any lost motion in the forward thrust may be eliminated. The n''t is kept in place by

the keeper *I*. That part of the gear casing which contains this thrust compensator also serves as the back bearing for the worm shaft. It is bushed or lined with bronze, and oil holes are provided for free access of oil from the reservoir which lubricates the worm and gear.

This device is but one example of the many contrivances of this nature and is sufficient to illustrate the method of taking care of the end thrust. Some makers use steel balls between the thrust plates with a view to reducing friction to a minimum. Fig. 160. There are some objections to the use of balls as they are liable to crush under heavy loads, the lost motion cannot be so easily taken up, and they wear grooves in the plates or rings. Some makers consider that the oil from the reservoir which lubricates the worm and gear is liable

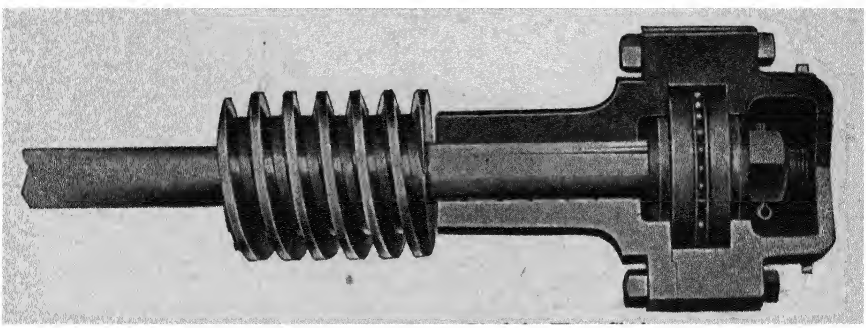


Fig. 160. Ball Bearing Thrust Block for Worm Shaft
*Courtesy of Warner Elevator and Manufacturing Company*

to contain grit and cuttings from the gear and, therefore, prefer to lubricate the thrust bearing by means of a special oil cup in the outside of the gear casing.

**Winding Drum and Cables.** Elevators which have a travel of less than one hundred feet seldom are made to run at a speed of more than two hundred and fifty f. p. m., and more frequently less than that. In the case of what is called the "drum" type of machine, the lifting and drum counterweight cables are attached to the drum, which is grooved spirally with a concave groove to receive and guide the cables. Each cable, or to be more accurate each pair of cables, has a separate groove, for there are but two grooves on the drum, and the cables are so attached and arranged that, while the lifting cables are being wound up, the counterpoise cables which are also attached

to the drum, are being unwound. A careful inspection of Fig. 150 will enable one to follow the two sets of cables to the car and counterpoise weights. The "idler" pulley slides from side to side as the cables wind on or off. It is seen, therefore, that in pairs they alternately use the same grooves. These grooves, or scores as they are called, are, in the case of the overhead type, so made that they run

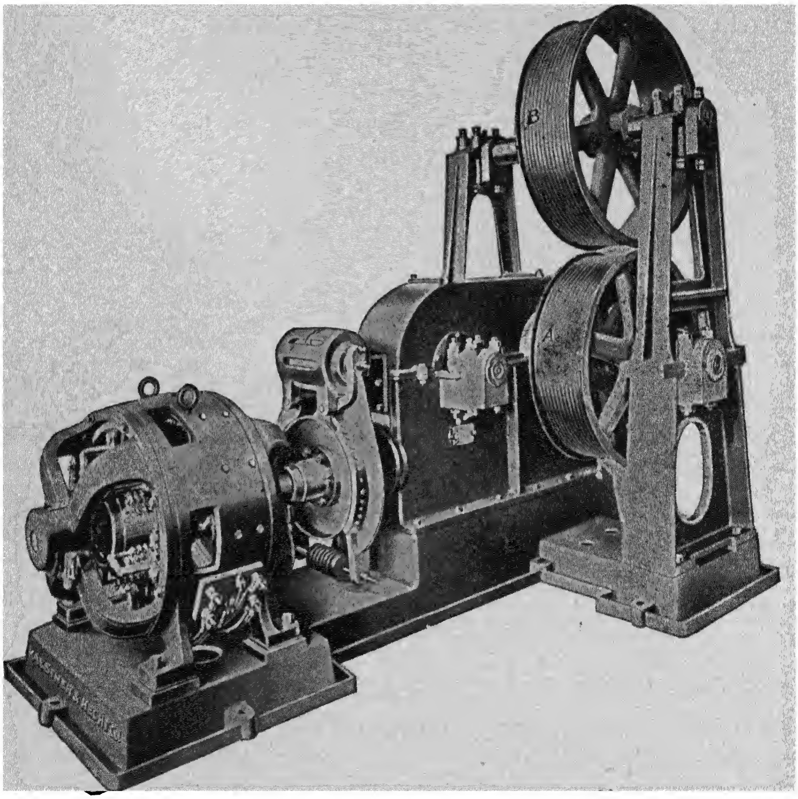


Fig. 161. Heavy Duty Tandem-Gear Traction Machine

from the ends of the drum towards the center, one groove on each side; when the engine is set to one side of the hatchway, either on a foundation or on suitable framing on one of the other floors of the building, the grooves run in pairs side by side from one end of the drum to the other and are made to lead right hand or left as the conditions require.

**Counterpoise Weights.** Two counterpoise weights are used: one which is attached to the drum to offset certain percentage of the

load to be lifted; and another used to counterbalance the car. The latter weight travels in the same runways as the drum counterweight, but above it, grooves or channels being cast in this upper weight to allow the supporting cables for the lower weight to pass through. The cables from this upper, or car, counterpoise pass up the hatchway and over sheaves or grooved wheels set at the top of the hatchway and thence down to the car itself, Fig. 150. It will be seen from the above description that in all six cables are used, and as the principal cables are rigidly attached at one end to the winding drum, the relation of the car to the number of revolutions of the drum is

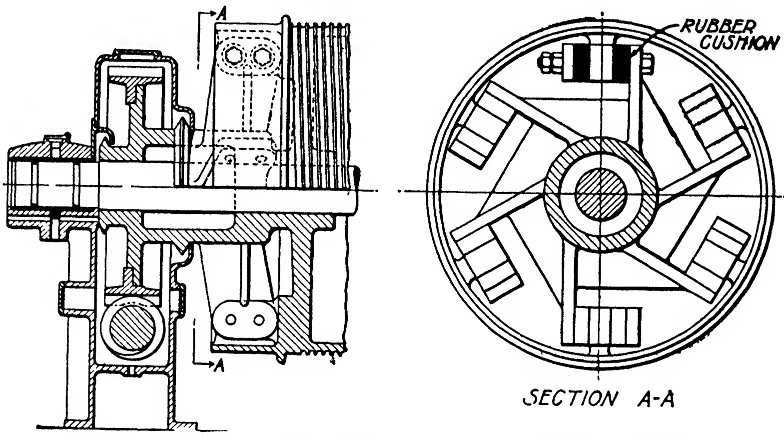



Fig. 162. Diagram Showing Six-Armed Yoke for Driving Traction Elevator

always fixed. It is evident, then, that a mechanical automatic limit stop, driven from the drum shaft itself, will always be sure to stop the engine at the end of the car travel.

## TRACTION ELEVATORS

The transmission previously described is designed for vertical lifts of one hundred feet or less. When the car travel is from one hundred twenty-five to two hundred fifty feet, new conditions prevail. *First,* the speed must be greater or the elevator will require too great a time to make a round trip and will, in consequence, prove inadequate for the service required; *second,* the drum face required to wind up so much cable will be too wide to be practical in an ordinary width of hatchway; and *third,* the weight of such extreme

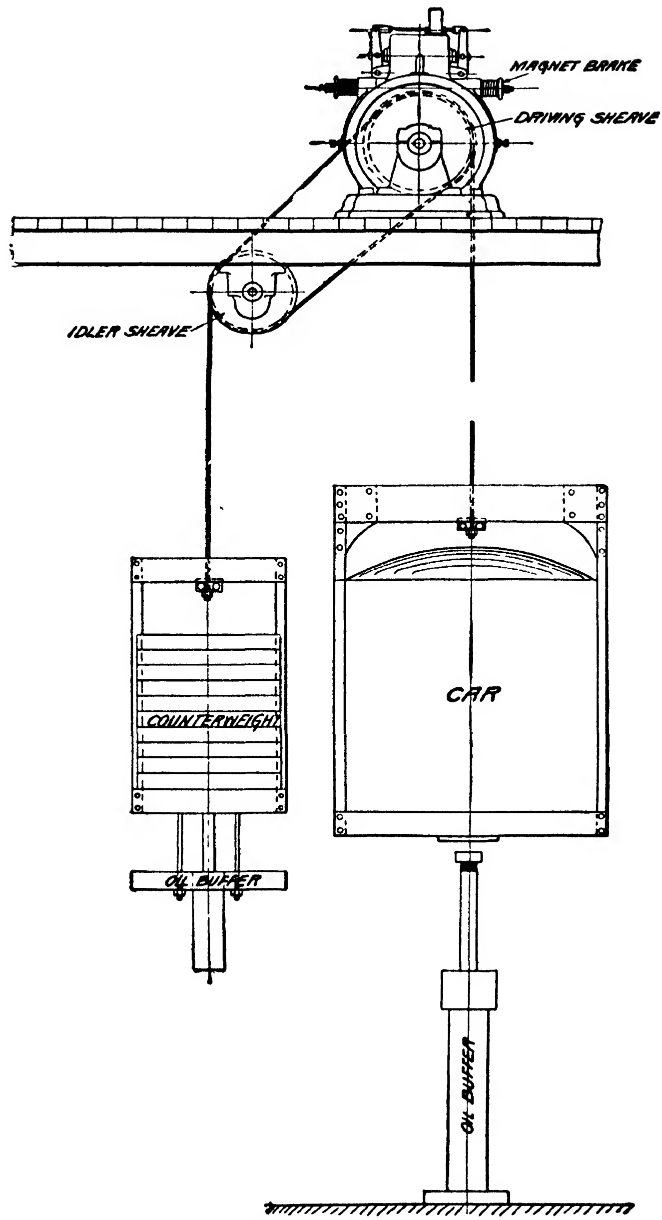



**Fig. 163. Diagram Showing General Arrangement of Roping for Traction Elevator**
*Courtesy of Otis Elevator Company*

lengths of cable will cause a series of vibration in the lifting capacity of the machine. To overcome these difficulties, another type of machine, called the traction elevator, has been devised.

**Drum and Cable Arrangement.** In engines of this type the wide faced drum is not used, being replaced by a drum not more than 12 inches wide, *A*, Fig. 161, with twelve grooves which are no longer spiral but are separate and distinct grooves such as are turned in a sheave; in fact, the drum is simply a sheave having more than the usual number of grooves. This drum is attached to or driven by the gear through a spider or six-armed yoke, Fig. 162, which engages with the arms of the drum. Rubber cushions are inserted at the points of contact to help soften any jar occurring at stopping and starting, and are held in position by suitable bolts and plates. Directly below or above this drum—according to whether it is an overhead engine or one arranged to be below on a foundation—is placed the "idler" sheave or drum *B*, Fig. 161, with the same number of straight grooves turned on its periphery, and running independently of the machine on a shaft and bearing of its own.

The cables are six in number and are passed over one of these drums and under the other in succession, Fig. 163. It is, however, always so arranged that they go twice partially around the engine drum and only once on the idler, although frequently, before leading into the hatchway, they are deflected by the idler in order to lead them plumb over the weight or over the car or into the hatchway, as the case may be. This accounts for the necessity of having the same number of grooves in each drum. One end of each of these cables is attached to the car and the other end to the counterpoise weight. It will be seen, therefore, that the cables are not attached to the drum at all, but depend on the friction with the face of the driving drum to transmit the power from the drum. Hence, the name *traction* machine. Figs. 161, 164, and 165, give a good idea of the machines of this type.

**Limit-Stop Arrangements.** *Hatch Switches.* With the arrangement of drum and cable just mentioned, the same width of face of drum will do for any height of car travel, but as the cables are not attached directly to the drum, the use of a mechanical limit stop is inexpedient, owing to a slight amount of slippage due to the stretching of the cables, quick starts and stops, and other causes. On this

account a limit stop which was actuated by the engine would not remain accurate for long. This uncertainty has been removed by

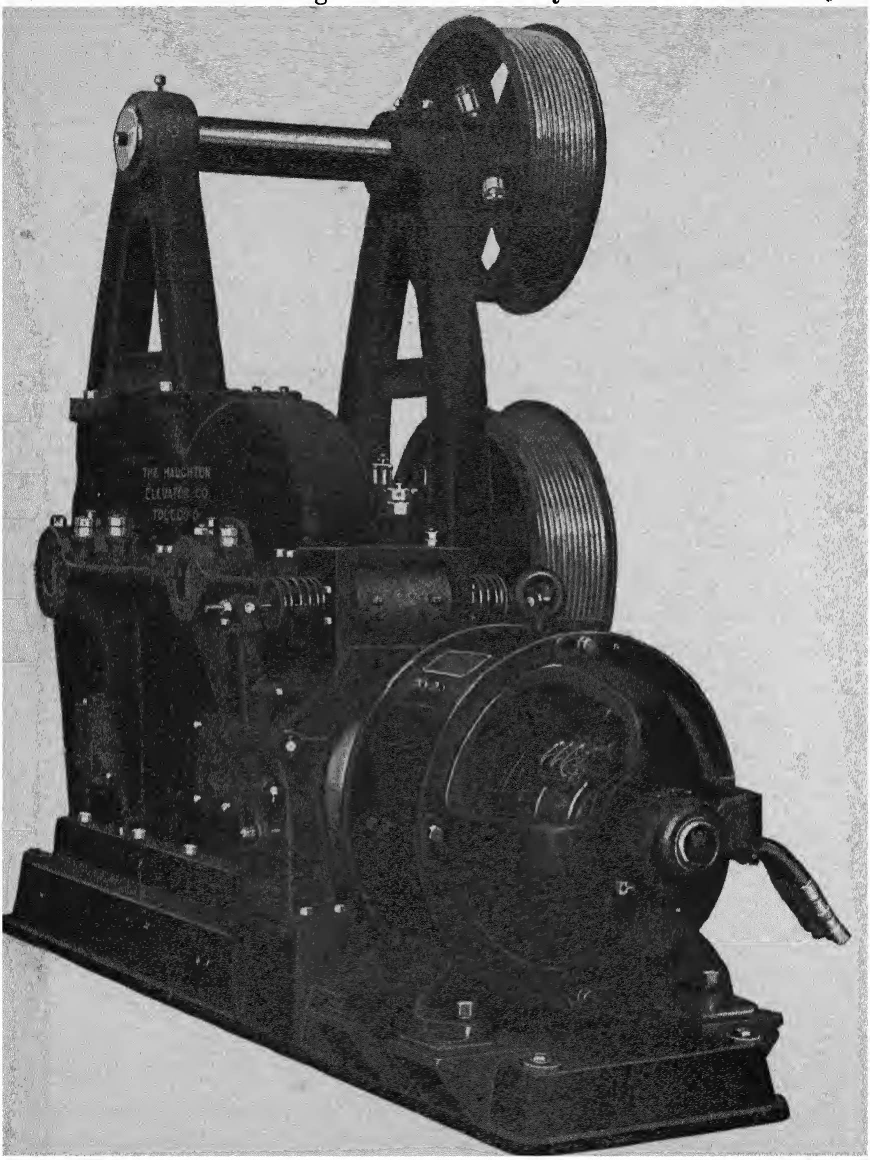


Fig. 164. Haughton Type of Tandem-Gear Traction Elevator
*Courtesy of Haughton Elevator and Machine Company*

the use of hatch switches, as shown at the left of Fig. 145. These switches are attached within the upper and lower limits of travel,

usually to one of the guides, so that the car in passing pushes back a lever and thereby opens or closes a circuit as required. For high speed three of these switches are used at each limit of travel: the first to slow down, the second to stop, and the third as an emergency switch, setting fairly close to the stop switch, which closes a circuit to a solenoid, thus applying an extra squeeze to the brake.

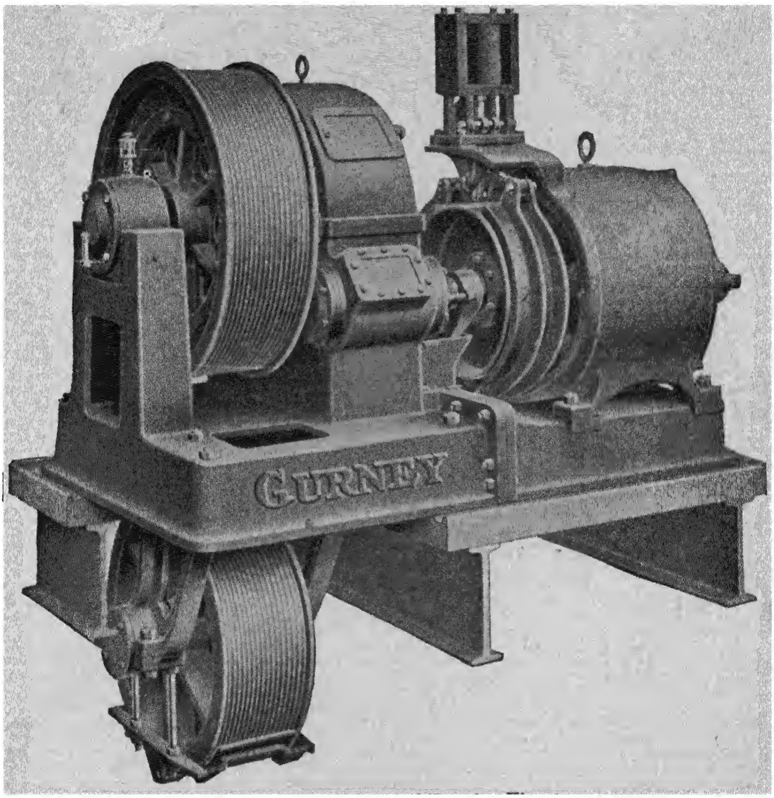



Fig. 165. Gurney Type of Traction Elevator
*Courtesy of Gurney Elevator Company*

*Oil Buffers.* In addition to the above safeguards, oil buffers are used below the car and the counterpoise weight, one type being shown in Fig. 166. The device consists of a plunger hanging vertically downward from the car and operating in a hollow cylinder filled with oil. A tank or reservoir of somewhat greater capacity than the cylinder takes care of the excess of oil when the plunger operates. If the car slips by the landing, the lower end of the cyl-

inder strikes a stop such as a pier or beam located at this point for the purpose, and the car or counterweight forces the plunger into the cylinder. The oil escapes into the tank through grooves cut in the surface of the cylinder, but as its passage through these grooves is necessarily slow, the car is slowly brought to rest.

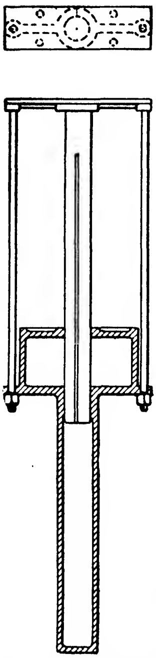


Fig. 166. Oil Buffer Attached to Car

In case the car is at the top landing when a slippage of this kind occurs, the counterpoise weight—which is at the bottom of its travel—performs the office by slackening the cables on the drum as soon as its descent is retarded by the oil buffer, thus destroying the tractive power of the drum for the time being.

Another type of oil buffer which is set in the bottom of the hatchway is shown in Fig. 167.

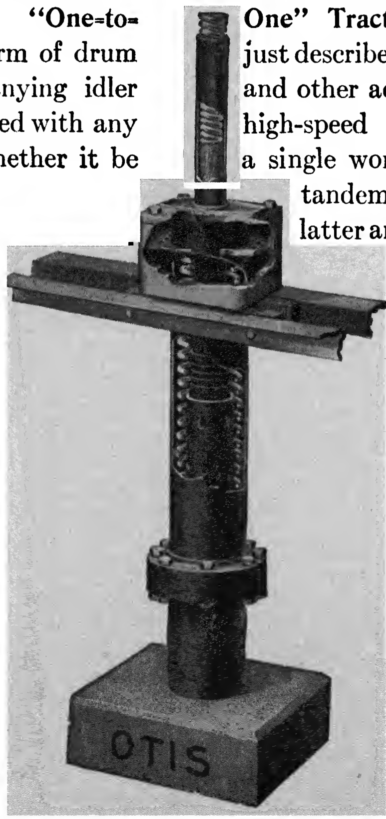



Fig. 167. Otis Oil Buffer with Spring Return

**"One-to-One" Traction Type.** The form of drum just described, with its accompanying idler and other accessories, may be used with any high-speed electric engine, whether it be a single worm and gear or a tandem machine. The latter are more frequently used, however, because the wear is divided between two worms and gears and the mechanism is less liable to heat or get out of order. These geared machines, however, are suitable only for moderately high speeds up to 400 f. p. m.; when a higher speed is desired, another form—known among elevator makers as the "one-to-one" engine—has been adopted. It derives its name from the fact that the drum performs one revolution to every revolution of the motor, the

drum being keyed directly on the armature shaft, which is extended for the purpose, and an additional or outside bearing provided for its support.

A machine of this type is shown in Fig. 168. A motor of special design is used, which is provided with windings for acceleration and retardation, and whose armature is wound for very slow speeds, viz, 50 r. p. m., and in many cases 35 r. p. m., for the full or normal

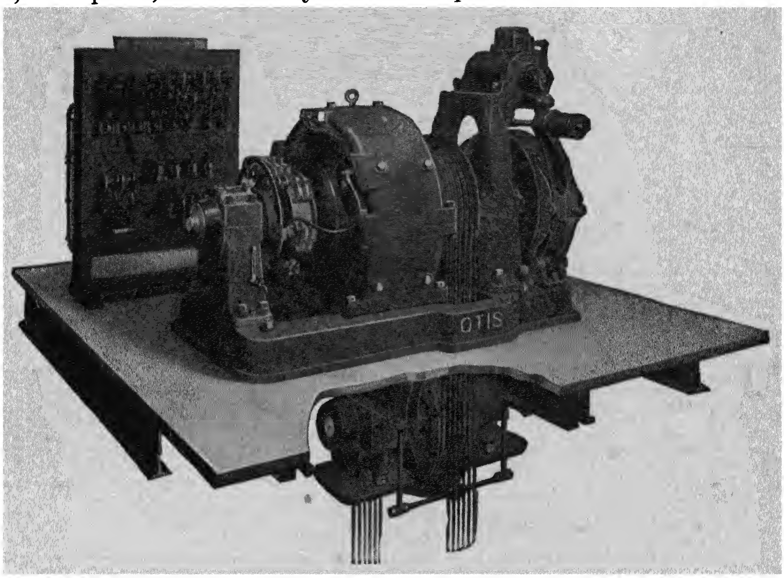



Fig. 168. Gearless Traction Elevator with "One-to-One" Roping
*Courtesy of Otis Elevator Company*

speed. A motor running at either of these speeds and a drum of, say, 34-inch diameter will produce a car speed of from 300 to 450 f. p. m., and with a little higher motor speed, say 60 to 70 r. p. m., a car speed of from 500 to 600 f. p. m. may be obtained.

When engines of this type are used, the armature shaft must be of ample diameter—5 inches to 6 inches according to the duty required of it—and the drum must be cast with a broad and heavy brake pulley integral with it. This pulley is of the same diameter as the drum, has a full 12-inch face, and has fitted to it a very powerful brake, applied in the usual manner by strong spiral springs and a very powerful solenoid for the release. The method of control of this elevator is similar to that used with the geared machine except that the motor is fitted with a speed governor driven from

the armature shaft, which is designed, in case of undue speed, to close a circuit to the solenoid, and at the same time to open the main circuit and thereby stop the engine.

**"Herringbone" Spur-Gear Type.** The latest type of electric elevator is what is called the "herringbone" spur-gear machine, Fig. 169. It consists of the usual bed plate with gear case, drum, bearings, and motor, mounted upon it, but instead of the worm and gear, a bronze spur gear and pinion are used to drive the drum. The ratio between gears is about 5 to 7 and the teeth in gear and

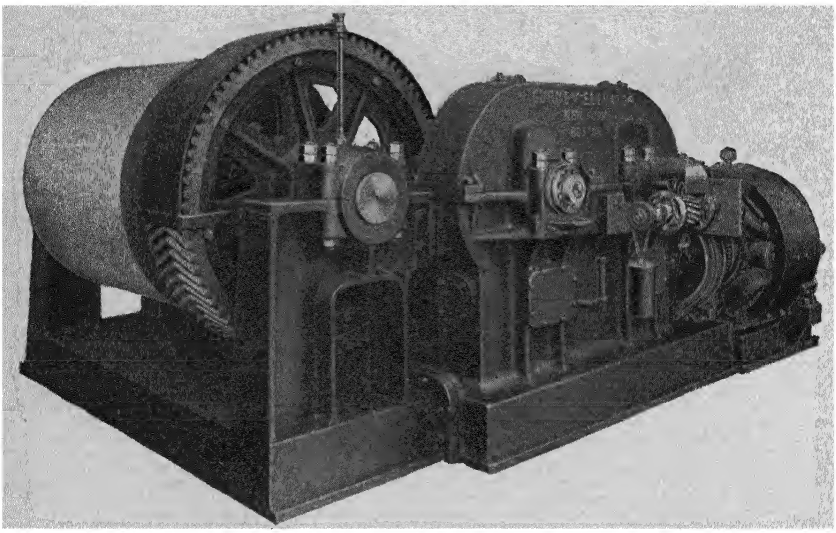


Fig. 169 Tandem-Gear Elevator with Herringbone Reduction Gearing
*Courtesy of Gurney Elevator Company*

pinion converge from the center of its face outward at an angle of about 60 degrees, Fig. 170; hence the name herringbone.

There is nothing new about this type of spur gear, as it has been in use for mill purposes for at least a century. It was originally designed to impart a smoother motion to the driven machines and also to give greater strength to the teeth, but until recently the only method of producing it was by casting and, of course, cast gears were not applicable to elevator service. However, cut gears of this description have now been successfully produced and they are being adopted in many other lines.

The advantages of the herringbone gear are a minimum of friction as compared with the worm and worm gear; smoother running than with the teeth cut straight across the face; and greater

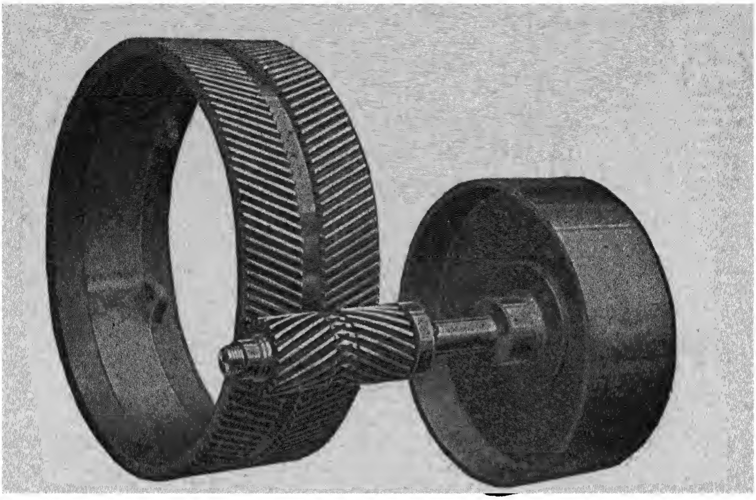


Fig. 170. Helical Gear Rim, Pinion, and Brake Pulley
*Courtesy of Gurney Elevator Company*

strength due to the diagonal position of the teeth, which allows a greater number of teeth to be in mesh with the pinion at one time.

The disadvantage, which is common to all spur gears, is its extreme liability to race when lowering a heavy load or when an empty car is ascending. To guard against this, the engine must be provided with a centrifugal governor, Fig. 171, on the pinion shaft, which applies a brake momentarily when a certain speed is exceeded.

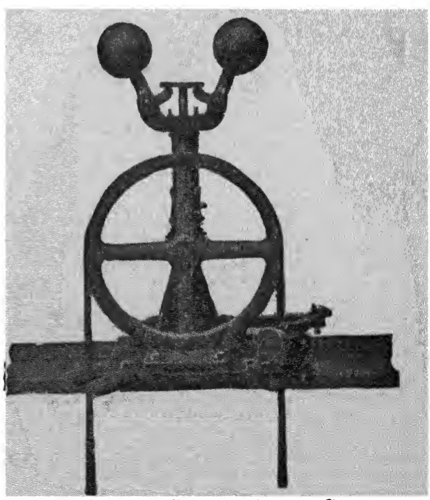


Fig. 171. Gurney Centrifugal Safety Governor Thrown into Operation

In this machine the motor is coupled to the pinion shaft to which the braking device is attached. The hoisting drum is provided with a heavy wide brake pulley such as is used on the

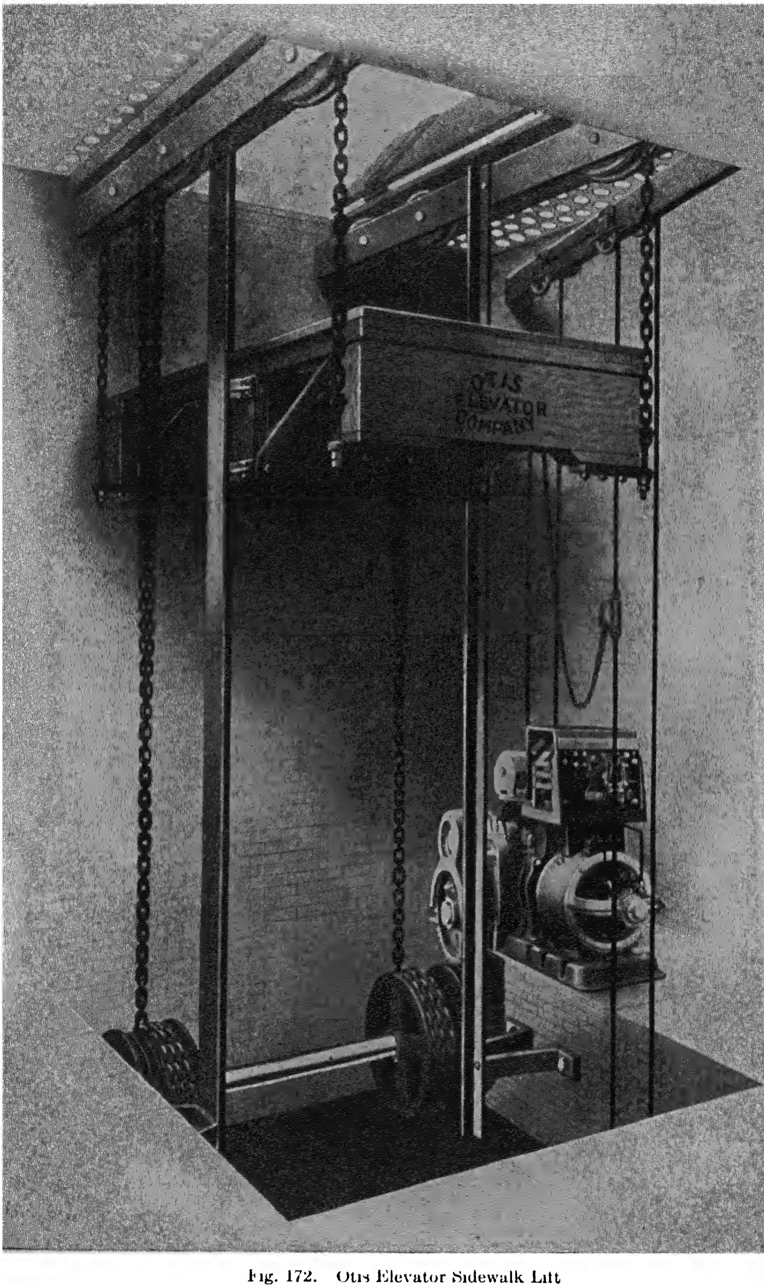



Fig. 172. Otis Elevator Sidewalk Lift
*Courtesy of Otis Elevator Company*

"one-to-one" type, and a powerful brake is provided which is used only in emergencies. This brake is applied by means of a lever and weight, which are held out of service by a latch or bolt arranged to be tripped by another centrifugal governor when a dangerous speed is attained. The motor used runs at a higher speed than that used with the "one-to-one" engine, but at a lower speed than that used with the worm and gear.

**Difficulties with Traction Types.** With all traction elevators, no matter whether driven through the medium of gearing or directly from the armature shaft, there is a danger of slippage of the cables on the driving drum, especially if the cables become greasy. This slippage is most noticeable when the operator endeavors to stop in descending with a heavy load. This is especially true if the speed is high and the attempt is made to stop quickly, the car sometimes sliding past the landing even to the extent of a story or two; should this occur (and it has done so in many instances) at the lower landing, a very unpleasant jar results.

The cause of this slippage is usually insufficient counterpoising, or if the building is at least twelve stories, it is probably due in a measure to the preponderance of weight on the car side of the drum, owing to the fact that the cables hang almost wholly in the hatch above the car. The remedy in such a case is the use of the chain counterpoise.

**Miscellaneous Elevators.** *Single-Belt Type.* There are other forms of electric elevators which are occasionally used. One of these, known as the single-belt elevator, consists of a worm-geared machine, hung from the ceiling of the room and adjacent to the hatchway, and driven by a motor through the medium of a leather belt. No countershaft is used, but the motor, which is reversible, is also located on, or hung from, the same ceiling at a distance of eight or nine feet from the winding gear, and drives it by means of the above mentioned belt. The motor is provided with a sliding base frame and screws in order to compensate for stretch of belt; the controller is attached to the frame or casing of the winding gear and is properly wired to the motor.

*Sidewalk Type.* Another type is the electric sidewalk or basement lift, Fig. 172. It is heavily built and is driven by a small electric engine.

**Mabbs Elevator.** The Mabbs type of electric elevator differs in one essential feature from all others previously described in that instead of being bolted securely to a concrete foundation in the basement or to steel beams overhead, the motor, with its gearing, framework, etc., travels vertically in guides moving in unison with the car it operates but in the opposite direction. Fig. 172A shows the general arrangement.

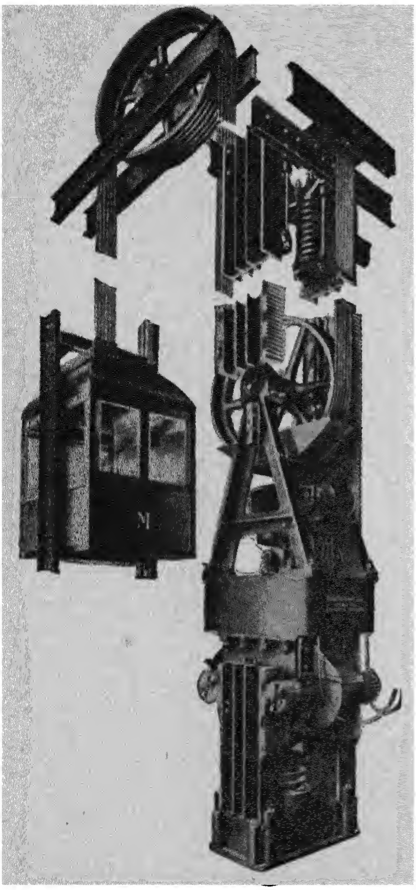
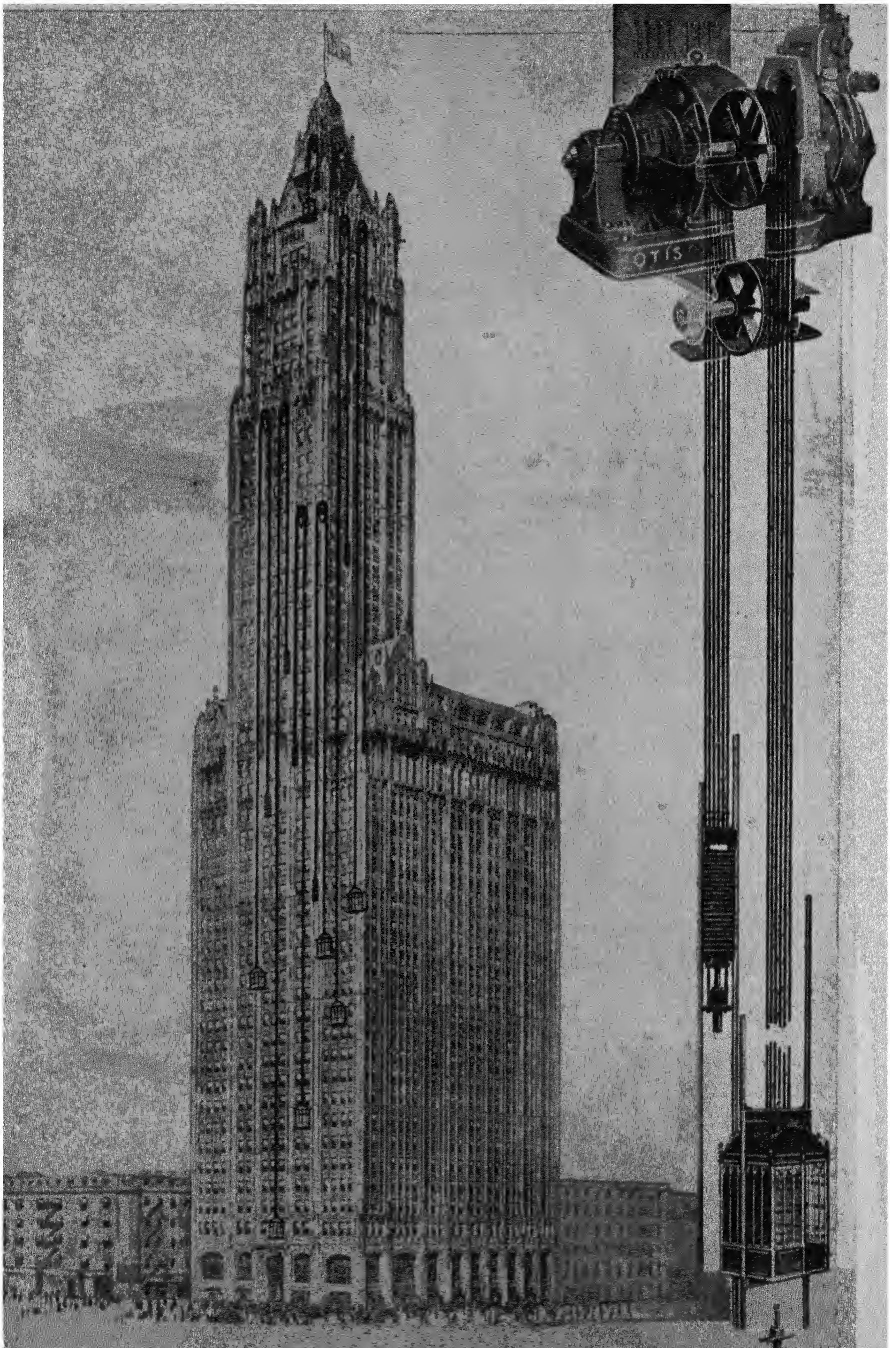


Fig. 172A. Mabbs Electrical Elevator

The armature shaft and worm are mounted vertically. The worm drives two worm wheels which operate four spur pinions on racks bolted to vertical cast-iron columns. A multiple-grooved sheave is set in a frame attached to and directly above the motor; this motor frame is connected to the car by means of a number of cables fastened to the stationary framework above the motor and running down to and underneath the sheave, thence up to and over the top sheave above the hatchway, and from there to the car. The motive part of this elevator has a purchase of two to one over the car (in addition to the reduction produced by the worm and spur gearing already mentioned). The motor, therefore, travels only half the distance the car does and at the same time acts as a counterpoise to the car.

Current is conveyed to the motor by means of copper-lined channels attached to the engine tracks. It seems to the writer that flexible electric cables would be more practical and economical. Several machines installed in Chicago in 1905 have proved very economical, but as designed are too expensive to build.




**ELEVATOR EQUIPMENT FOR THE WOOLWORTH BUILDING, NEW YORK CITY, AND THE TYPE OF ELEVATOR USED**

*Courtesy of Otis Elevator Company, New York City*

# ELEVATORS

## PART IV

## EQUIPMENT DESIGN AND CONSTRUCTION

### PROPORTIONING SIMPLE PARTS

**General Procedure.** In designing any kind of machine there are two operations to be considered which are almost distinct from each other. The procedure consists in determining (1) the general principles upon which the work to be performed depends for its proper fulfillment, and (2) the exact arrangement of the details and their proper proportions to produce a harmonious and reliable operation. It is with the latter feature that this chapter deals, and in doing so it is necessary not only to estimate the amount of stress on the part or member under consideration, but also to ascertain exactly the way in which the stress acts.

### APPLICATION OF PRINCIPLES

**Action of Stresses.** The stresses to which elevator machinery is subjected may be classed under two heads: (1) pressure without motion, or static stress, and (2) pressure with motion, or dynamic stress. The first is simple to deal with because its amount generally can be ascertained and usually is not variable. If the stress is in the nature of tension, it represents a force tending to tear the member asunder, and if in the nature of compression, it represents the force tending to shorten or to compress the member. If it is a cross stress, the office of the member under stress is that of a beam. In the case of dynamic stress the intensity often varies, and frequently tends to set up and to maintain a vibratory motion, which is sometimes intermittent and at other times continuous and increasing in intensity during the motion of the machine. Accordingly, it is very important that the designer should become familiar with these features beforehand in order to so proportion the various parts of the machine that they may not only withstand the pressure to which

they are subjected, but also do it without showing any appreciable effect of the stress to which they are subjected.

The action of stresses on the members of a machine being so varied, it is necessary to be acquainted with the properties of the materials entering into the construction of the machine. Where sudden and violent strains are to be anticipated, some material should be used which, in addition to the requisite strength, possesses sufficient elasticity to have the property of diminishing the maximum effect of the force to which it is subjected by yielding to a certain degree at the moment of impact; but it should not yield beyond its elastic limit, for that would impair its safety. The designer shows his skill in becoming conversant with the nature and extent of these different forces and in so proportioning the various members of a machine that they will withstand them properly and safely. Parts under a compressive stress must be so proportioned as to stand up under the load without crushing or bending.

**Factors of Safety.** In order to safeguard against undue and unforeseen strains, and also against unseen imperfections in materials or workmanship, it is customary in estimating the dimensions of each part to make the part from 4 to 10 times as strong as the theoretical size based upon the known strength of the material of which it is composed and the stress to which it is to be subjected. This multiple is called the *factor of safety,* and is denoted by the symbol $f$ in the following examples.

In elevator designing the factors of safety vary largely, according to the nature of the duties to be performed. For moving parts it is from 6 to 10, according to the amount of wear and the liability to breakage through impact or sudden and severe strains. For standing parts a factor of 5 is usually allowed. Where and why these varying factors of safety are applied will be fully explained as the subject is discussed, it being assumed that the student is familiar with the essential properties of various materials for use in the construction of these machines.

## EMPIRICAL DESIGN

**Proportions of Hoisting Hook.** The hook used with the hoisting rope of the sling machine, Fig. 1, discussed in Part I of Elevators, will be taken up first. The proportions of such a hook, as used for

hand-operated elevators, are indicated in Fig. 173; (*A*) showing the relative sizes of the sections at *xx* and *yy*, and (*B*) showing the proportions of the heaviest section of the hook *yy* enlarged. These hooks usually are made from some commercial size of round bar iron or mild steel for convenience and economy in manufacture. The cross-section (*B*) shows that the shape of the part of the hook under greatest bending moment approximates an oval, with one portion larger than the other. The larger part of this section is the size of the bar stock from which the hook is made, while the thickness of the smaller part should be ½ the diameter of the bar, and the length through the oval 3 times the radius of the bar.

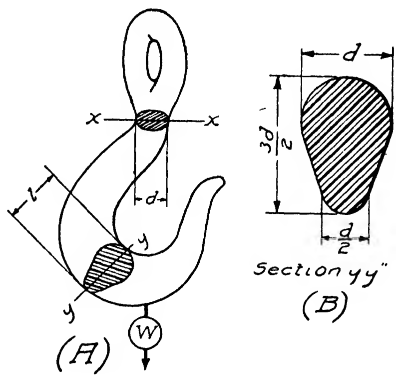


Fig 173. Diagram of Hook Proportions

*Rule for Load.* When of these proportions, simply square the depth or length $l$ of the oval in inches to find the safe load in *tons* $W$ for a sling hook. The simple equation for this is $W=l^2$, where $l=\frac{3d}{2}$, $d$ being the thickness of the bar stock in inches. Conversely, to find the proper depth of section to carry a given load, find the square root of the load $W$ in tons, and the result is the proper depth $l$ in inches, i.e., $l=\sqrt{W}$.

**Size of Gudgeons.** For gudgeons, Fig. 174, projecting from the ends of log drums — Fig. 2, Part I of Elevators — and for journals at the ends of drum shafts of the more recent styles of hand elevators, it has been customary to use a journal $1\frac{3}{8}$ to $1\frac{1}{2}$ inches in diameter. This size, however, was determined on at a time when a knowledge of the proper proportions was not so thorough as it is now. Still, it will be found upon examination that these sizes are

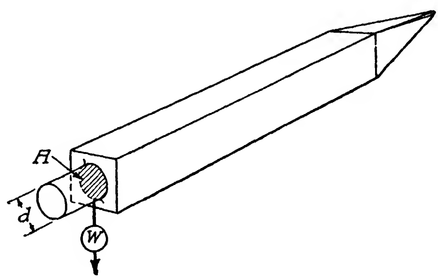


Fig. 174. Detail of Gudgeon

not very far out of the way, and the writer can say from actual observation, extending over a period of more than 25 years, that he has never yet seen one of the gudgeons broken in the ordinary usage to which they are subjected.

*Rule for Determining Diameter.* The rule for determining the size of these gudgeons is to allow 1 square inch of sectional area for each ton of the load in *tons* $W$ to be carried; or $A=W$. To find $d$ the diameter of the section having this area $A$, of which the relation is $A=\pi\left(\frac{d}{2}\right)^2$, divide the tonnage $W$, to which corresponds the number of square inches of required sectional area $A$, by .785; and extract the square root of the quotient. The result will be the diameter $d$ in inches; the equation being $d=\sqrt{\frac{W}{.785}}$.

*Illustrative Examples. 1.* Consider the heaviest load usually placed on the platform of an ordinary hand-power elevator, which is not more than 3000 pounds — $W$ equals $1\frac{1}{2}$ tons. According to the rule, the required sectional area $A$ would be 1.5 square inches. This number divided by .785 equals 1.91, the square root of which is 1.38, which is the diameter $d$. This is very close to $1\frac{3}{8}$ inches, the usual diameter of the gudgeon.

*2.* Suppose the load $W$ to be 2 tons. Then 2 divided by .785 equals 2.55, and $d$, the square root of 2.55, is 1.59, which is only slightly more than a diameter of $1\frac{1}{2}$ inches.

## DESIGN OF DRUM SHAFTS

### Hand-Power Elevator Type

**Influence of Drum.** *Diameter of Drum.* Let the plain shaft on which is mounted the drum of a hand-power elevator, Fig. 175, be discussed for a suspended load of 3000 pounds. The drum used for a machine of this capacity is found by experience to give the best results if of 16-inch diameter. If it were made according to the rule laid down by the makers of wire rope — drum diameter is 40 times the thickness of the rope, or $D=40t$ — it would have to be 20 inches in diameter. If it were of this size, however, it would necessitate such large gears and shafts that the resultant machine would be very cumbersome and unwieldy, and, moreover, much more costly.

*Effect of Wooden Barrel.* With a slow running machine, such as a hand-power elevator, it has been found in practice that by using drums of which the barrel, or that part over which the cables wind, is made of wood — Fig. 11, in Part I of Elevators — the cables do not suffer so much as they would if running at a higher speed in a sheave. Hence, the use of wooden drums is customary. Furthermore, a cable also suffers less when running over a drum of somewhat smaller diameter than when running in a sheave, because there is less stress on the parts around the drum and little or no slippage. This depends, however, upon whether the cables have their ends fast, or grip the drum by traction. In the latter case, if there is any slippage, the wood barrel, being softer than the cables, wears instead of the cables.

**Determination of Shaft Diameter.** To find the proper diameter for a Bessemer-steel or a wrought-iron shaft for a drum to wind up a given load, use should be made of one of two values of an extreme fiber-stress or loading constant $k$ — 20,500 for Bessemer steel, and 12,000 for wrought iron. The diameter can then be found from the following rule.

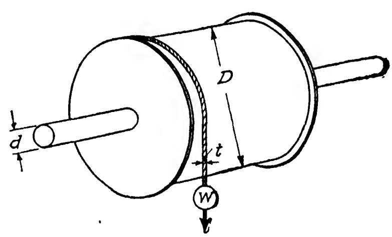


Fig. 175. Diagram of Short Drum and Shaft

*Rule for Shaft in Torsion.* Multiply the load to be raised in *pounds* $W$ by the factor of safety $f$ desired, and by the radius of the drum $\frac{1}{2}D$; divide the result by the appropriate loading constant $k$; and extract the cube root of the quotient. The result is the proper diameter $d$ of the shaft in inches; the equation being $d = \sqrt[3]{\frac{Wf\frac{1}{2}D}{k}}$.

*Illustrative Example.* In the previous problems the load $W$ to be raised by hand-power elevator was 3000 pounds and the diameter $D$ of drum 16 inches. It is now desired to find the proper diameter $d$ of a plain Bessemer-steel shaft for this drum and load.

Very few parts of an elevator are subjected to more severe stresses than the drum shaft. The starting under load, the sudden stopping, and other stresses due to loading and unloading the car, at times all cause severe stress on this member of the machine. Therefore, it is necessary to use a high factor of safety, partly for this

reason and partly because it is a moving part, or one that is in motion frequently. So the factor of safety in this case may be taken as 10. Multiplying the load, 3000 pounds, by the factor of safety 10, and by 8, the radius of the drum in inches, the result is 240,000. Dividing this result by the constant 20,500 gives 11.7, which in this case may be called 12. The cube root of 12 is 2.29, which is the calculated diameter of the shaft. In actual practice the shafts for this capacity of machine are made $2\frac{7}{16}$ inches in diameter.

*Effect of Loading.* From motives of economy some builders make the drum shafts $2\frac{5}{16}$ inches, and this size does fairly well, but $2\frac{7}{16}$ inches gives better results. This might be expected when it is remembered that the gear in a hand-power elevator is frequently located nearly a foot distant from the drum. This distance increases the tendency toward torsion. In some cases the shaft is extended beyond the hoisting drum for a distance of 2 feet or more for the reception of a drum which carries the counterweight. This weight, although seldom exceeding 600 or 700 pounds, exerts a tendency to help the shaft in winding up the load. To further sustain the shaft a center bearing is used between the drums to support it under the transverse stresses due to loading and unloading.

### Drum Shafts for Two-Belted Machines

**Conditions in Proportioning.** *Drum Size.* Apply the rule for shaft diameter to the case of a two-belted worm-geared elevator, similar to that in Fig. 42, Part I of Elevators. As the usual capacity of one of these machines is $1\frac{1}{2}$ tons, the same load as in the previous case may be used as a basis. In this style of elevator the diameter of drum used is generally 30 inches. It seldom varies from this because it is a convenient size amply large in diameter for the cables, which in a two-belt machine of this capacity are usually $\frac{3}{4}$ inch in thickness, and occasionally $\frac{5}{8}$ inch. The required speed is always obtained by making the pulley on the driving shaft of a diameter suitable for that speed, which is usually 50 feet per minute.

*Allowance for All Factors.* It must be remembered that before making any calculation to determine the proper size of any member of a machine a careful study of all the conditions surrounding its service and duties must be made, for if by neglect, ignorance, or inadvertence, any omission is made, the results arrived at will be erroneous

and perhaps lead to disaster or failure. Consequently, it is imperative for a designer to be certain of the accuracy of his premises before proceeding to make calculations.

*Effect of Counterpoise.* In the present case the load to be lifted is 3000 pounds, but there is also the cage or car which is counterpoised, sometimes from the drum and sometimes from the cage itself. In the latter arrangement the car cannot be fully counterpoised, for, if it were, it would not descend. In fact, in any two-belt elevator the car is seldom evenly counterpoised, except in cases where it is driven from a countershaft by a gas engine or motor, and there is then the extra weight of the counterpoise hanging from the drum.

**Calculation of Shaft Size.** *Load Including Unbalanced Weight.* Take the case where the unbalanced weight of the car, usually about 500 pounds for this type, is to be made to overhaul the weight and cables for descending when it is at the top story. This extra weight, or unbalanced weight of the car, always has to be lifted by the drum and must be added to the live load of 3000 pounds, making the effective load 3500 pounds.

*Application of Formula.* Under these conditions, applying the rule already given for diameter of shaft in torsion, we have — by multiplying the load 3500 by the factor of safety 10 and by the radius of the drum, 15 inches, and dividing by 20,500, the fiber-stress constant for steel — a result of 25, the cube root of which is 2.92. Hence a 3-inch shaft would be used in practice. This is the size generally used for this type of machine, but allowances are made for keyseats by making the body of the shaft about $3\frac{3}{16}$ inches where the drum is keyed on and also where the gear is attached, inasmuch as the keyseats cut away a portion of the shaft where they are located.

### Drum Shafts for Belted Electric Elevators

**Affecting Conditions.** *Over-counterweighting.* In applying the previous rule to the shaft of the drum of a single-belt electric machine of the same capacity, the conditions surrounding it must first be considered. Such a machine is shown in Fig. 185, but we will consider a case where the load is carried on the drum as in Fig. 186, and as illustrated at (*B*) in diagram (*II*), Fig. 184. In this machine it is customary to over-counterweight, for the sake of economizing power for operation — that is, to add counterweight, not only to the extent

of the weight of the car, but an additional weight as high as 30 or 40 per cent of the live load to be lifted. While the addition of this extra counterpoise adds to the efficiency of the machine, and at the same time lightens the load on the teeth of the gear and the threads of the worm, thereby dividing up the wear between both sides of the teeth and threads, it imposes heavier burdens on the drum shaft.

*Transverse-Stress Factor.* This brings another factor under consideration, and that is whether the shaft, aside from its capacity to wind up, will be strong enough to bear safely both the counterpoise and the load. It must be remembered that the shaft will not have to withstand the torsion of winding up at one time all this extra counterpoise weight and the load as well, because they are attached to the drum from opposite sides, and therefore in a measure counteract each other, as designed. It is not, then, the torsional stress which must be considered, but the stress due to carrying a live load with the shaft acting as a beam — that is, a transverse stress.

**Determination of Shaft Size.** *Loading Conditions.* Let the load on the drum shaft now be considered. First, there is the live load of 3000 pounds to be lifted; then the weight of the car, which is, say 1400 pounds; then the weight of 1400 pounds to counterpoise the car; and, finally, there is an over-counterpoise of about 30 per cent of the live load to be lifted, say 1000 pounds; making a total static load of 6800 pounds.

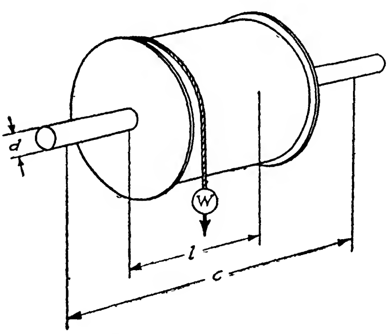


Fig. 176. Diagram of Long Drum and Shaft

*Bearing Spacing.* The drum still is assumed to be 30 inches in diameter, as for the previous type — although that does not enter into this calculation except as to distance $c$ between bearings or the length of the shaft — and this diameter is now considered in order to obtain the width $l$ of the face of the drum, Fig. 176. Let it be assumed that the elevator is for a five-story building, and that 2 cables are used for hoisting and 2 for the counterpoise. With the requisite laps or turns at each end, the drum would be of about 24-inch face, and for that width it should have two sets of arms, one at each end, with a hub, say 6 inches long, for

each set; so that the distance from where the drum takes a bearing on the shaft to the journal boxes in which the shaft revolves will be not more than 3 inches at each end, or $c$ minus $l$ equals 6 inches in all.

*Procedure for Shaft under Transverse Stress.* Now, under these conditions, to find the proper diameter of shaft to sustain a load, a procedure similar to that used before is employed, except that 14,400, a different value of the loading constant $k$, is used as a divisor. In order that the student may clearly understand the procedure employed, the rule for finding the proper diameter of the shaft for a transverse stress when the bearings are not close together, referring to Fig. 176, is given as follows:

*Rule for Shaft Diameter with Bearings Well Apart.* Multiply the load $W$ in pounds by $f$ the factor of safety required, and by the net effective distance $c'$ — or $c-l$ — between the bearings in inches; and divide the product by the constant $k$, which equals 14,400; the cube root of the quotient is the proper diameter $d$ of the shaft in inches. Stated as an equation this is $d=\sqrt[3]{\frac{Wfc'}{k}}$.

*Illustrative Examples. 1.* The load on the shaft under consideration is concentrated at 3 inches from each bearing, which is practically 6 inches between bearings. The total static load is 6800 pounds; the factor of safety should be 10, the same as for the torsional strain; making the estimated load 68,000 pounds. This multiplied by 6 equals 408,000, which divided by 14,400 equals 28.3. The cube root of 28.3 is slightly over 3, showing that so far the drum shaft used for the two-belt elevator would be strong enough.

But suppose the drum required were of only 12-inch face; one set of arms would be enough, and, while the distance between the outer bearing and the gearcase bearing would be less, the net effective distance would be greater. With one set of arms, the hub should be longer, both to resist side stress when the cables are wound all at one end of the drum, and also to give greater section in the one key and more bearing for it.

*2.* For a drum of 12-inch face and with a single set of arms the hub should not be less than 8 inches long. If a space of 3 inches were allowed on each side between the end of the drum and the bearing, the distance on each side from bearing to hub end would be 5 inches, or 10 inches in all. This is the usual custom. Now,

multiplying the load of 6800 pounds by the factor of safety 10, and by the distance 10 inches between bearings, gives 680,000, which divided by 14,400 gives 47. The cube root of 47 is 3.6.

On the whole, it seems better, therefore, to make the shaft for a single-belt electric elevator somewhat larger than would be used on a two-belt worm-gear machine, on account of the extra heavy counterpoise. This is what is done in practice, the shaft being $3\frac{7}{16}$ inches in diameter for work similar to that described here.

### Drum Shafts for Direct Connected Electric Elevators

**Loading Conditions.** *Drum Carrying Torsional Load.* The general rule also applies to direct connected electric elevators where the conditions are the same, Fig. 186, but in most of the better class of this type the gear is bolted to a flange, which is in one piece with the drum and connected thereto by a cast-iron neck forming an extension of one hub of the drum, Fig. 162, Part III of Elevators, and Fig. 180. With this construction the torsional stress is taken care of entirely by the drum itself, and the only stress on the drum shaft is transverse.

*Shaft Torsion in High-Speed Traction Machine.* In determining the proper diameter of shaft for the drum of a traction machine, let us consider the same load of 3000 pounds, as before, so that comparisons may easily be drawn. Here the conditions surrounding the service differ somewhat. Instead of the slowly moving car traveling from 50 to 150 feet per minute, Figs. 185 and 186, the speed is from 300 to 600 feet per minute, Figs. 206, 207, and 254, and the horsepower required is accordingly greater, being from 35 to 70. While the drum shaft in a measure is relieved from the stresses accruing from the application of the shoe brake, on account of the brake pulley being a part of the drum itself, there are other stresses resulting from the application of the dynamic brake and from the acceleration of speed at starting which are fully as intense, if not more so, than those resulting from the shoe brake. Therefore, it is very necessary to be sure that all conditions are carefully considered before applying the rule to find the diameter of shaft.

The stopping of a load running at a speed of 600 feet per minute is a very different matter from that of stopping the same load when running at from 60 to 150 feet per minute. The acceleration of the

same load from a very low linear velocity of say about 60 feet per minute to one of 600 feet per minute within a distance of from 10 to 15 feet is almost as great a strain on the drum shaft. Both these operations are performed through the medium of this shaft, and this extra service has to be provided for. The stress is not equivalent to that of an equal load dropping through the same distance, but it has been proved by tests to be fully equal to $\frac{1}{3}$ of that. Accordingly, to allow for the rapid acceleration or retardation in this high-speed type, practice is to multiply the actual load by 5, which, combined with the ordinary factor of safety, is the effective factor of safety $f'$.

**Size of Traction-Machine Shaft.** *Rule for Shaft in Torsion.* It is customary to counterpoise a traction machine to the extent of the weight of the cage and cab and 25 per cent of the full load lifted, and therefore the unbalanced or effective load to be lifted is 75 per cent of the live load $W$ actually on the car. Accordingly, the rule for the diameter of a traction-machine shaft, considering torsion, is $d=\sqrt[3]{\frac{.75Wf'\frac{1}{2}D}{k}}$.

*Illustrative Example.* Applying this rule to a steel shaft when the elevator capacity is 3000 pounds, the effective net live load to be lifted is 2250 pounds. Multiplying the effective load by 5 for the reasons previously stated, and again by the ordinary factor of safety 10, gives 112,500. Multiplying this by 15 the radius of the 30-inch drum, and dividing by the safe fiber-stress constant for steel, 20,500, the result is 82. The cube root of 82 is 4.34 inches, which is the diameter of the traction-machine drum shaft estimated for torsional resistance with high acceleration.

*Rule for Checking Shaft Size.* This result may be checked by using another rule, which employs one of two values of a different constant $k'$ — 82 for steel shafts, and 110 for iron shafts — and is as follows: Multiply the appropriate constant $k'$ by $P$, the horsepower required; and divide this product by the number of revolutions per minute $n$; the cube root of the quotient is the diameter $d$ of the shaft. This may be represented by $d=\sqrt[3]{\frac{k'P}{n}}$.

*Illustrative Example.* Now, by the rule for horsepower given subsequently under the section on Power Requirements—

$\frac{\text{foot pounds per minute}}{33{,}000}$ — it is found that the theoretical horsepower in lifting 3000 pounds 600 feet per minute is 54.5. Supplementing this, the only additions that have to be made, owing to the absence of any gearing between the armature shaft and the load, are to take care of the cable loss, which in cases like this usually is figured at 10 per cent, and the loss in the motor, which is about 15 per cent. The allowance for these two losses is all that is added to the theoretical horsepower, so that adding 25 per cent of 54.5, say 14, a total of 68.5 horsepower is required, and a 75-horsepower motor may be selected.

The shaft considered is to be of steel and capable of full load at maximum speed. For a linear velocity of 600 feet per minute a 30-inch drum makes about 75 revolutions per minute. Applying the rule, the constant 82 times 75 horsepower divided by 75 revolutions per minute equals 82, the cube root of which is 4.34 inches, the requisite diameter of shaft to withstand the torsional strain.

But there is also a transverse strain due to the static load which must be considered. This comprises the load to be lifted, 3000 pounds; the weight of the car and cab, say 3500 pounds, on one side of drum; and the counterpoise weight on the other side of the drum, equaling the weight of the car and cab and say 30 per cent of load on 4500 pounds. This makes a total load of 11,000 pounds. Now by the rule given on page 227, supposing the bearings to be two feet apart:

$$11{,}000\times 10\times 24\div 14{,}400=183$$

The cube root of this is 5.67 inches. Considering this as a diameter, its area will be 25.24 square inches. Taking also the diameter obtained by computing the torsional strain, 5.34 inches, its area is 22.39 square inches. Adding these together we have a total of 47.64 square inches; the diameter corresponding to this area is about $7\frac{3}{8}$ inches, which would be the proper diameter of a shaft for this service.

As this is the minimum diameter to meet these conditions any portion of the shaft which is to contain keyseats must be left larger.

## TRANSMISSION PARTS

### PROPORTIONING OF GEARING

**Principal Considerations.** In selecting gears for use in lifting loads, the first point to be determined is the reduction ratio or relative leverage required. This is usually governed to some extent by

other conditions, such as the speed and type of the prime mover — whether it is an engine, an electric motor, a line shaft, or other means — the required speed of the elevator car, and the most convenient size of drum. The size of the drum, however, is governed also to some extent by the load to be lifted, which determines as well the size of the cables or wire ropes used. These latter features are explained subsequently in this section under their proper heading. What is now desired is to explain the method of determining the proper pitch or spacing of the teeth and the strength of the arms and rim of the gear.

## Pitch of Gear

**Determination of Pitch.** The determining of the proper thickness of tooth, or, what is the same thing, the proper pitch to carry a given load safely, is a matter that is generally conceded to be a difficult and uncertain problem, and most writers on the subject dismiss it after saying, in addition to that statement, that it must be largely based on experience in actual practice. The following rule is one which has been used for years in actual work and will be found reliable, simple, and easily remembered; but close attention must be paid to seeing that all the details are complete before applying it — the same may be said of any rule. Beginners frequently commit the error of omitting some trifling but very essential detail, the absence of which leads them into difficulties.

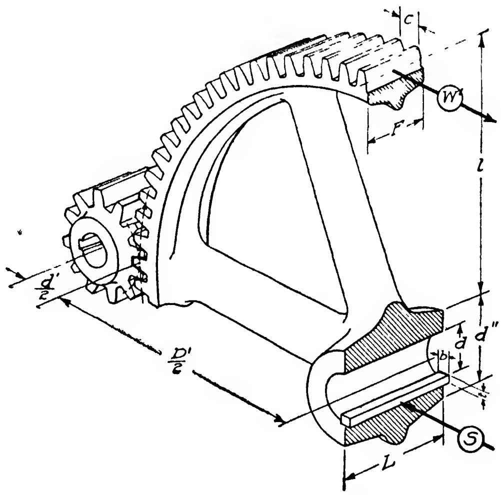



Fig. 177. Diagram of Spur Gear Proportions

**Pitch of Cast-Iron Spur Gears.** The rule referred to is for cast-iron spur gears with the teeth cast in shape, Fig. 177, and applies only to sound castings made of strong gray iron where $F$, the face of

the teeth, is made not less than $2\frac{1}{2}$ times the pitch $c$ — or $F=2\frac{1}{2}\,c$ — the rule being as follows:

*Rule for Spur-Gear Pitch.* Multiply the square root of the load $W'$ in pounds at the pitch line of the gear by .03; and the product is the pitch $c$ in inches and decimals of an inch. The equation for the pitch, then, is $c=.03\sqrt{W'}$.

*Illustrative Examples. 1.* For example, it is desired to construct a handpower elevator to lift 3000 pounds. What pitch should be used for the gear, and what diameters and face for the gear and the pinion?

It is taken for granted that the student has read the section on Hand-Power Elevators — Part I of Elevators — and has learned from it that the usual diameter of drum for this capacity is 16 inches, while that of the hand-rope wheel is 5 feet, or 60 inches; also, that the ratio or purchase on a lift of this capacity should be 28 or 30 to 1. The first thing to do is to find the ratio between the drum and the rope wheel, which is 60 inches divided by 16 inches, or 3.75. A ratio of 7 to 1 for gear and pinion — $\frac{D'}{d'}$, Fig. 177 — would make a total purchase between the hand rope and lifting cables of $3.75\times 7$, or 26.25, which is close enough for all purposes.

(*a*) It will be seen by reference to the description of the hand elevator that the diameter of the gear used is usually about double that of the drum. For convenience try a gear having a pitch diameter $D'$ of 30 inches, and see what the load $W'$ will be on the rim of the gear at the pitch line. This load is proportional to the load $W$ on the drum in the inverse ratio of the diameters $D'$ of the gear and $D$ of the drum, since the couples $W\times D$ and $W'\times D'$ must equal each other; so $W'=W\frac{D}{D'}$. Thus, by the relation $W':3000::16:30$, or $W'=3000\times\frac{16}{30}$, $W'$ is found to be equal to 1600 pounds, which represents the load at the pitch line of the gear. The square root of 1600 is 40, which multiplied by .03 gives the pitch of the spur gear as 1.20 inches. The pitch of the gear will be sufficient if it is $1\frac{1}{8}$ inches, and it might be made $1\frac{1}{4}$ inches, but this would necessitate using a gear of larger diameter. It will be found by computation that a spur gear of 30.24 inches pitch diameter will contain just 84 teeth at $1\frac{1}{8}$-inch pitch. This quantity divided by the reduction ratio 7 gives 12 teeth for the pinion, which is small enough.

(*b*) Trying, now, a 36-inch gear, it is found from the relation $W'=3000\times\frac{16}{36}$ that the load $W'$ on the pitch line is 1333 pounds. The square root of 1333 is 36.5, which multiplied by .03 equals 1.095, or nearly 1.1 inches. A pitch of $1\frac{1}{8}$ inches is the nearest to this, and for a spur gear 36 inches in diameter it would give 100 teeth, with 14 teeth for the pinion. Either of these combinations would do very well.

*2.* As another example, suppose the load is 2000 pounds and the drum is 16 inches in diameter. A gear 24 inches in diameter would be suitable for this. Then $W'=2000\times\frac{16}{24}$, giving 1333 pounds as the value of load $W'$ on the pitch line, which is the same as was obtained for the 36-inch gear with a 3000-pound load on the drum. Of course, the same pitch will do. As before, the square root of 1333 is 36.5, which multiplied by .03 equals 1.095. A 1-inch pitch is frequently used for this size or capacity of machine, but is rather light.

**Pitch of Cast-Iron Worm Gears.** *Calculation by Spur-Gear Rule.* Suppose a worm-gear belt-power or electric elevator is to be built having a drum of diameter $D$ 30 inches, and a gear of diameter $D'$ about 27 inches; what ought to be the pitch to carry safely a load $W$ of 6000 pounds? Here the load $W'$ on the teeth of the gear is greater than at the periphery of the drum, the gear being smaller in diameter, so that the relation $W'$:6000::30:27, or $W'=6000\times\frac{30}{27}$, gives 6666 pounds as the value of $W'$. The square root of this is 81.65, which multiplied by .03 gives 2.45 inches as the proper pitch for a cast-iron gear, according to the previous rule.

*Effect of Distributed Stress.* Here the conditions differ; the gear is propelled by a worm which always has four teeth of the gear in mesh — Fig. 50, Part I of Elevators — and moreover the bottom of the tooth is concave. The latter of these features adds strength, and the other distributes the load, making the total stress on any one tooth much less than in the case of a spur gear. Hence, the pitch may be reduced at least 25 per cent for any worm gear of cast iron with a concave face and of such diameter that four teeth are in mesh with the worm at the same time. This would make the proper pitch $1\frac{3}{4}$ inches for cast iron, and for bronze it may be reduced to $1\frac{1}{2}$ inches with perfect safety.

*Rule Modified for Worm-Gear Pitch.* In the case of worm gears of cast metal, multiply the pitch obtained according to the rule for

spur gears by a constant $k''$ — .73 for cast-iron, and .63 for bronze gears — to obtain the right proportion in pitch $c$; the relation being $c=k''.03\sqrt{W'}$.

**Design of Heavy-Duty Gears.** *Worm- and Spur-Gear Combinations.* One more example will be given to illustrate another deviation from the general rule, and this, it is believed, will cover all the exceptions. Suppose an exceptionally heavy machine is to be built — say of 10,000 pounds capacity — such as is not called for frequently, and for which it would not pay to make an entire set of patterns. Use could be made of a worm and gear suitable for a 5000- or 6000-pound load, and, instead of putting a drum on the worm-gear shaft, a spur or a helical pinion could be used, as in Fig. 169, Elevators, Part III, making this pinion drive a gear attached to the drum.

For convenience the spur gear could be the same diameter as the drum, so as not to project too far into the hatchway, and also to avoid the necessity for driving the worm gear too fast. The ratio of gear and pinion could be either 2 to 1 or 3 to 1, according to the size of the worm gear used. Of course, some additions in the way of stands for the drum shaft and an extension of the bed for them to rest on would be necessary.

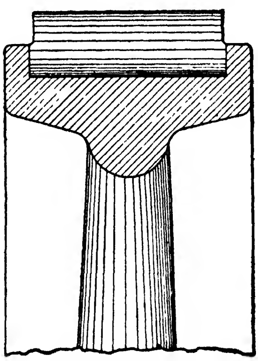


Fig. 178. Shrouded Spur Gear

*Duplex Arrangement of Spur Gears.* As suggested, the spur gear has a pitch diameter $D'$ the same as the diameter $D$ of the drum, so the load $W'$ at the pitch line would be the same as $W$, namely, 10,000 pounds. Hence, the square root of this multiplied by .03 gives 3 inches as the required pitch. A single gear, however, at one end of the drum, especially if it is a long drum — that is, if the lift is high — would necessitate heavy arms and rim to resist the torsional stress. Therefore, a better arrangement would be to put a spur gear at each end of the drum, also making the worm-gear shaft longer and using two spur pinions. The face and pitch of each spur gear and pinion then could be correspondingly reduced, making them of $1\frac{3}{4}$- or 2-inch pitch and about 5-inch face.

*Use of Shrouded Gears.* When additional strength in the teeth is desired, they can be shrouded as high as the pitch line on both gear

and pinion, Fig. 178, stock being left to turn these shrouds down exactly to the pitch line. When assembled in the machine the gear and pinion could be set so that the peripheries of the shrouds would roll on one another. This arrangement prevents the gears from meshing too deeply and results in a more smoothly running machine.

### Design of Gear Parts

**Proportions of Worm Gears.** It must be remembered that whenever a spur gear is figured by the rule given the face or length of tooth of the gear must be at least $2\frac{1}{2}$ times the pitch; but in figuring the worm-gear face $F$ the proper relation is $\frac{2}{3}$ of the diameter $d'$ of the worm — $F=\frac{2}{3}d'$ — or, since the diameter $d'$ should not exceed $\frac{1}{8}$ the diameter $D'$ of the worm gear, the face $F$ may be .08 $D'$ in such a case. The proportions of the worm are shown in Part I of Elevators in the section on Worm and Gear, illustrated by Figs. 43 and 49 to 53.

**Proportions of Spur Teeth.** For cast spur gears, Fig. 177, the teeth should be in the following proportions:

| | |
|---|---|
| Pitch equals | 1.00 |
| Length of face | 2.50 |
| Depth of tooth | .75 |
| Working depth or mesh | .70 |
| Clearance | .05 |
| Thickness of tooth | .45 |
| Width of space | .55 |
| Play | .10 |
| Height beyond pitch line | .35 |

**Design of Gear Rim.** The thickness of the rim at the outside edges of the cast-metal spur-tooth gear is usually made slightly more than $\frac{1}{2}$ the pitch, and tapers or thickens in toward the center of the rim on the inside or where the arms begin. Sometimes a rib is run around inside the rim in the center where the arms join the rim.

*Computation for Hand-Power Machine.* The rim of the 30-inch gear for the case of hand-power elevator is computed as follows. The face of the gear is to be not less than $2\frac{1}{2}$ times the $1\frac{1}{8}$-inch pitch, or $2\frac{13}{16}$ inches; so, for the purpose of simplifying the calculations, asssume a 3-inch face. Let the thickness of rim at the outer edge be $\frac{9}{16}$ inches, the thickness at the center of the rim 1 inch, the rib being

of $1\frac{1}{4}$ inches over-all depth and $\frac{3}{4}$ inch wide, making the section of the rim of the shape shown in Fig. 177. Under these conditions the rim as shown will be found to have an area of about 3 inches, and taking 13,000 pounds as the allowable stress per square inch — for reasons discussed later in the section on the Use of Cast Iron — the tensile strength of the rim will be equal to 39,000 pounds. As $W'$, the load on the gear teeth, is 1600 pounds in the case considered, the margin of safety is nearly 25, which is sufficient to cover possible defects in casting and other weaknesses.

**Design of Gear Arms.** *Cantilever Function.* The arms of the gear may be considered as cantilevers, when estimating the strength required for them, the fixed end being the portion which joins the hub of the gear, and the loaded end being that which joins the rim.

The hub, in all cases where it is keyed to the shaft to transmit power to it or to receive power from it, should have a diameter $d''$, Fig. 177, at least twice the diameter $d$ of the shaft — or $d''=2d$ — so for the $2\frac{7}{16}$-inch shaft of the case considered the hub should be 5 inches in diameter. With a 30-inch gear, then, the distance $l$ from the hub to the rim should be 1 foot. This is the length of the arm, so the arm is equivalent to a lever firmly imbedded in some stable foundation at one end, and projecting 1 foot at the other end upon which a load $W'$ of 1600 pounds is carried. Hence, it is necessary to calculate what should be the depth of the beam or lever where it enters the part to which it is attached or in which it is embedded.

*Rule for Size.* The coefficient of strength, or, in other words, the safe resisting power of cast iron when under transverse strain, is placed by most authorities at 5544 pounds per square inch of perpendicular section, that is, in the direction of the stress. In a cantilever it is customary to multiply the load by 12 as a factor of safety. Accordingly, using the symbols in connection with Fig. 177, the area of the arm base section may be found from the relation $A=\frac{W'fl}{5544}$, in which $A$ is the area of the section required, $W'$ is the load applied at the gear teeth, $f$ is the factor of safety, and $l$ is the length of the lever arm from where it joins the hub to the pitch circle where the load is applied.

*Illustrative Example.* For the case in question the arm section would have an area equal to the product of the load at the outer end

multiplied by 12 times 1 foot, the length of arm, and divided by the constant for cast iron, 5544. Thus, $\frac{1600 \times 12 \times 1}{5544} = 3.46$ square inches, the sectional area of the arm at the root. This arm, therefore, may be 1 inch thick and $3\frac{1}{2}$ inches wide where it joins the hub.

But since, of the six of these arms, no one is at any time under the entire stress of the load, the results may be, and are, modified to some extent. If of cruciform shape, a section $\frac{7}{8}$ inch thick by 3 inches wide at the base, with a rib on each side $\frac{1}{2}$ inch thick, will be found satisfactory. This rib does not add materially to the strength of the arm in the direction of the stress, but it tends to stiffen it sideways. If made oval in section, it would be $1\frac{1}{4}$ inches thick and 3 inches wide at the root. In either case it would taper to $2\frac{1}{4}$ or $2\frac{1}{2}$ inches wide at the rim. In the case of the oval arm it would be made 1 inch thick at the outer end and gradually taper to $1\frac{1}{4}$ inches thick at the base.

**Calculation of Hub Keys.** The length of the hub $L$, Fig. 177, is governed by the bearing surface on the shaft necessary for stability and by the shearing stress on the key. The proper width of key for a $2\frac{7}{16}$-inch shaft is $\frac{5}{8}$ inch, according to the standard adopted and used by manufacturers and engineers throughout the United States and Great Britain.

*Rule for Proportions.* To find the total shearing stress $S$ on any key in pounds — using the symbols in connection with Fig. 177 — multiply $W'$, the load in pounds at the periphery of the wheel, by $D'$, the diameter of the gear wheel in inches; and divide by $d$, the diameter of the shaft in inches. This may be stated as $S = W'\frac{D'}{d}$. The requisite cross-sectional area $A$ to resist shear is found by dividing the product of the total shearing stress $S$ and the factor of safety $f$ by the unit ultimate shearing stress $s$ for the material used; or $A = \frac{fS}{s}$. The proportions then are governed by the limiting dimension — the practice for the breadth $b$ of the key being about $\frac{1}{4}d$, and the thickness $t$ equal to $\frac{1}{2}b$ — so that the length $L$ would be $\frac{A}{b}$, or $L = \frac{fS}{sb}$.

*Illustrative Example.* When the load is 1600 pounds, and the diameter of the gear 30.25 inches, and that of the shaft $2\frac{7}{16}$ inches,

the stress on the key is 1600 times 30.25 divided by 2.4375, or 19,856 pounds. The ultimate shearing stress of mild steel is 36,000 pounds to the square inch. Using a factor of 5, it is found that the required shearing area is 19,856 times 5 divided by 36,000, or 2.76 square inches. The key being $\frac{5}{8}$ inch wide should therefore be 4.4 inches long, or in practice $4\frac{1}{2}$ inches long for the hub, in order to allow ample area for the key. This length would give sufficient stability for the hub, as it is $1\frac{1}{2}$ inches longer than the 3-inch face of the gear already decided upon, and there is no side stress.

### Use of Cast Iron

**Effect of Chemical Elements on Strength.** *Carbon, Silicon, and Phosphorus.* The tensile strength of cast iron is a very uncertain factor, varying greatly with the chemical constituents of the iron. An excess of carbon, while giving it increased strength, makes it too hard to work and increases its brittleness. Silicon up to 3 per cent softens cast iron; beyond that it makes it hard and adds nothing to its strength, but, on the contrary, makes it slightly weaker and gives it a tendency to shrink irregularly when cooling. This action causes shrink holes inside in the thicker parts which cool last, and, as these shrink holes cannot be seen, they are not suspected until fracture or boring or drilling reveals them. Phosphorus makes weak iron, giving it a tendency to break unexpectedly; its presence, even in minute quantities, is an undesirable feature.

*Sulphur.* Sulphur makes iron *hot short* — liable to break easily when hot — but does not affect it after it has become cold, its action being just the opposite of that of phosphorus. The bad feature of sulphur in iron, however, is that it causes shrinkage cracks, which occur in castings whose irregular thickness causes uneven cooling so that the part which cools last is liable to tear away from the other portion. On the other hand, where no sulphur is present but where there is silica, there will be a shrinkage hole inside the thick part instead of the crack on the outside.

*Mixture for Machinery Castings.* Chemical analysis of the pig iron before use in the foundry reveals all these defects, and there is no excuse for their existence in machinery castings. The best machinery cast iron contains about 4 per cent carbon and from 1 to $1\frac{1}{2}$ per cent silicon, no phosphorus, and no sulphur. Sometimes a

small percentage of manganese may be present, but opinions differ as to whether it is of any benefit or not, although it does no harm if not in excess.

**Care Necessary in Use.** In addition to what has been said it may be noted that not enough care always is taken to prevent the presence of loose dirt in the mold, and in skimming the ladle while pouring the casting. Neglect of these precautions results in unsound castings through the presence of foreign matter where there should be iron only.

*Allowance for Weakness.* These conditions added to the low tensile strength of cast iron make it necessary for the designing engineer to be very liberal in specifying the amount of metal to be used where a tensile or a transverse stress is to occur in cast iron. The tensile stress of cast iron is usually considered as 13,000 pounds to the square inch, and though some set it at 15,000 pounds the conservative engineer prefers to be safe, that is, to allow a safe margin.

## WINDING DRUMS

**Design of Arms and Hub.** Practically the same rules and methods which are given for gears apply to the drums used for winding up the hoisting and counterpoise cables, so far as they apply to the arms and hubs of the drums, and except that where the face of the drum is over 14 inches wide it is customary to use two sets of arms and two hubs for the sake of stability, Fig. 179. If there were only one set of arms and one hub for a drum with a wide face, there would be a strong tendency to tip when the cables were all wound up on one end of the drum, and this strain would be alternately at the opposite ends, so that eventually the drum would work loose on the shaft if the side strains did not in the meantime break the arms.

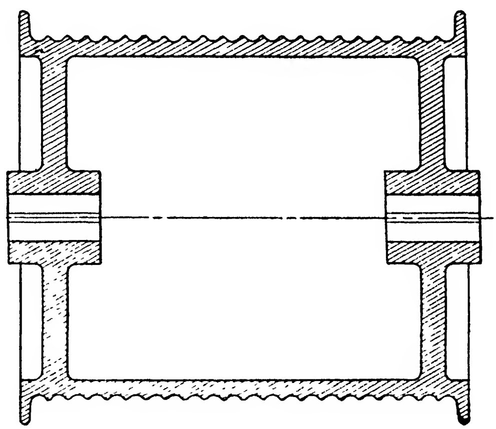


Fig. 179. Wide Drum with Double Hubs

**Drum Extension for Annular Gear.** When driven by a worm gear the drum frequently is made with a long neck and flange at one end, Fig. 162 — Elevators, Part III — and Fig. 180, the flange being used as the cast-iron center to which to bolt the bronze worm gear. In such case the gear casing is made with a large opening in one side to permit the introduction of the neck, and is formed with a trough or tray to catch the oil which may run down from the upper side of the

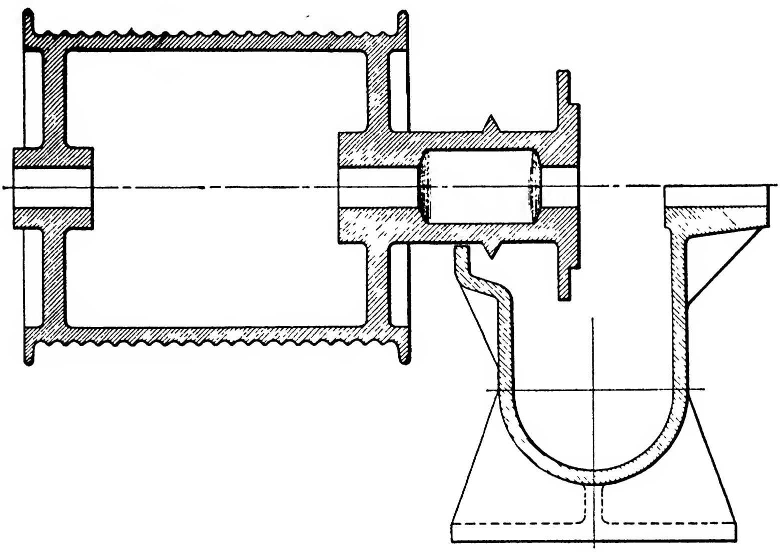


Fig 180. Drum with Integral Extension to Carry Gear

worm gear. The neck has a V-shaped ridge all around it to prevent the escape of the oil.

**Designing of Drum Barrel.** It has been found in actual practice that, for drums not exceeding 48 inches in diameter and for loads up to 3 tons, $\frac{5}{8}$ inch is thick enough for the barrel of the drum after turning and before scoring for the cables. This scoring is the turning of a spiral groove from one end to the other of the space traveled on the drum by the cable, to serve as a path to guide the cable as it winds onto the drum and to keep the different turns in regular order on the drum.

*Scores.* The score is usually $\frac{3}{16}$ inch in depth, and is made concave to fit the contour of the cable which is to run in it. Being a spiral, it forms one continuous path as far as it extends, and is always

cut of such a pitch as to leave $\frac{1}{16}$ inch between the different turns of cable. For example, for a $\frac{5}{8}$-inch cable the score would be made with a tool the cutting edge of which would be circular in form and of $\frac{5}{8}$-inch diameter, and the pitch would be $\frac{11}{16}$ inch. For a $\frac{3}{4}$-inch cable the tool would have a circular edge of $\frac{3}{4}$-inch diameter, and the pitch would be $\frac{13}{16}$ inches. When two cables are used two scores are cut side by side, and in such case the pitch is double; for $\frac{5}{8}$-inch cables it would be $1\frac{3}{8}$ inches, and for $\frac{3}{4}$-inch cables it would be $1\frac{5}{8}$ inches.

Sometimes the scores, instead of running side by side, start from either end of the drum, one being a right-hand and the other a left-hand spiral, and lead toward one another until they meet in the center of the face of the drum. This is most frequently the practice when the drum is located overhead, as in the case of the overhead electric, and also with the hand machine. Where the hoisting engine is located on a foundation alongside the hatchway it is usually found more convenient to run the cables in scores which parallel each other as first described. Under certain conditions a right-hand score, and under others a left-hand score, is found preferable. Where the load is 10,000 pounds or over the thickness of the drum face must be increased to 1 inch, or even to $1\frac{1}{4}$ inches under some conditions.

*Face Flanges.* The flanges located at the ends of the drum face to prevent loose cables from slipping over the ends are seldom made thicker than $\frac{3}{4}$ inch, and more frequently $\frac{1}{2}$ inch. They also help to strengthen the rim just at that part where the holes are usually bored through for fastening the ends of the wire cables.

## SHEAVE PULLEYS

**Action of Members.** In designing sheaves it must be remembered that the upper half of the rim of the sheave, Fig. 181, when in service, is an arch without abutments. The arms have to act alternately as pillars and as tie rods. It is comparatively easy, however, to design a sheave which, as far as the individual strength of each member is concerned, is amply able to perform its duties.

**Production Defects.** The process of making the sheave is such that it is surrounded by dangers which not only imperil it during production, but frequently even after it has been turned out of the foundry, an apparently sound casting, thus leaving it in a condition which is very likely to cause its failure at any time during

service. It is only by close attention to all the requirements of designing, molding, and casting that these troubles may be avoided with any degree of certainty.

*Shrinkage Strains.* Shrinkage strains are the most common trouble of the elevator maker so far as sheaves are concerned. If a sheave were made and proportioned exactly as it would figure out according to the rules for defining the strength of every member, the chances are that the casting would come out of the foundry with one or more shrinkage cracks, which would unfit it for use. Hence, the only safe way is first to figure out the proportions of every part according to the service required of it, and then to examine the design carefully to see if there are any parts so thin that they may cool much sooner than the rest and thereby cause flaws from shrinkage. These defects show themselves more frequently in the rim and at those places where the arms join the rim. This is due to the rim cooling first, and to the hub, which is much thicker, continuing to cool long after the rim and arms have set, thus tending to draw the arms away from the rim.

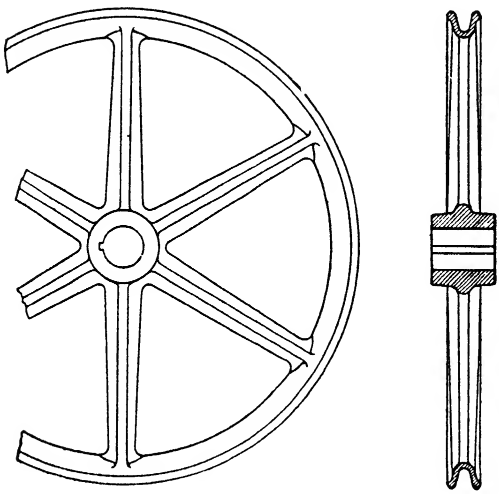


Fig. 181. Sheave Pulley

**Proportions of Sheave Members.** *Use of Bushings in Hubs.* Fortunately the sheave does not have to convey power to the shaft which serves as the axis on which it revolves, and hence the hub need not be made as heavy as for a gear. If the sheave is to run loose on its shaft, in all probability it will be bushed, that is, have its hub lined with a brass sleeve $\frac{1}{4}$ to $\frac{5}{16}$ inch thick. In this case the core for the casting may be $\frac{1}{2}$ inch larger in diameter, which is certainly a help, and the metal forming the hub need not be thicker than 1 to $1\frac{1}{8}$ inches, which is also a safeguard against causing internal strains in cooling.

*Extra Stock in Arms and Rim.* In making the design it is safe to say that the arms and rim, the latter especially, should be fully as heavy as those for a gear of equal diameter, not because they require it for strength, but to produce a sound casting. For the same reason it is best to cast the rim plain and to turn the grooves afterward. The cost of the extra metal in the rim is amply repaid in the certainty of getting a strong sound sheave. The arms of all sheaves should be strengthened against side strains by ribs of ample proportions on each side, and where the arms join the rim large fillets should be used as a means to prevent shrinkage cracks developing in the corners — in fact, corners of all kinds should be avoided.

*Scores in Set Sheave.* The scores or grooves for the cables to lie in must be deeper in the rim of the sheave than in the drum. Their depth in a set sheave — one having no lateral travel on the shaft — should be fully $\frac{1}{2}$ the diameter of the cable, and they should be turned an exact fit, with the corners of the grooves rounded slightly, so that, in case the cable is led sidewise at a slight angle, no corners are present to chafe the cable.

*Scores in Vibrating Sheave.* In the case of idler sheaves which travel from one end of a shaft to the other — vibrating sheaves, as they are frequently called — the groove or score for the cables or cable should have a depth of fully the diameter of cable used, and the score should be flaring, that is, wider at its mouth than at the bottom.

These sheaves are used mostly to lead the cables away from the drum in another direction, and, as the cables wind and unwind on and off the face of the drum, the sheave travels with the cable along the face of the drum in addition to its revolutions on the shaft — from which comes the name vibrating sheave. The travel endwise on the shaft often amounts to from 24 to 30 inches, and the sheave is caused to perform this lateral motion largely through the cable pressing against the sides of the score. The necessity for a deep flaring score and good ribs on the arms to resist side pressure will be readily seen.

*Hub of Vibrating Sheave.* A hub slightly longer than usual also is a help, especially after the bore of the sheave becomes somewhat worn, as the long hub not only retards the wear, but, when it does occur, helps to prevent the sheave from moving to one side and locking itself. These idler sheaves should always be bushed with a brass sleeve which can be removed and replaced by a new one when worn.

### Sheave-Pulley Shafts

**Size of Shaft.** The proper way to determine the size of shaft for either an idler or an overhead sheave has been given in previous pages in arriving at the size of drum shafts to sustain a load, but it is best to repeat it here because there is a supplementary rule which goes with it for use when the distance between the end bearings of the shaft is great, as it is in the case of vibrating sheaves.

*Rule for Shaft in Shear.* Referring to Fig. 182, the simple rule — as stated in connection with Gudgeons in a preceding section on Empirical Design — is to allow 1 square inch of sectional area for each ton of the reaction $R$ resulting from the total load $w+w'$ carried on the shaft; and to divide this quantity corresponding to the number of square inches of required sectional area $A$ by .785; and then to extract the square root of the quotient; the result being the diameter $d$ of the shaft in inches. With $A=R$, in *tons* — $R$ being the resultant load reaction, Fig. 184, at the bearing — the equation for the diameter is $d=\sqrt{\frac{R}{.785}}$.

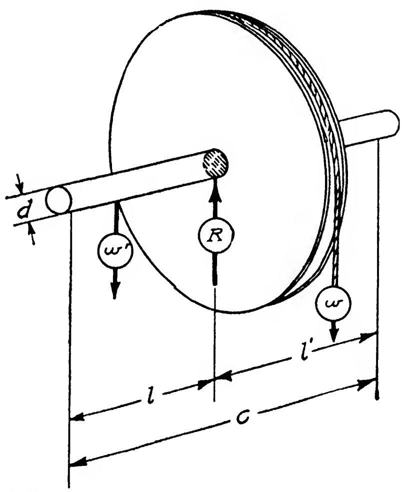


Fig 182. Sheave and Shaft Loading Diagram

This rule is for journals where the hub of the sheave is close to the box or bearing, and the stress on the shaft is consequently a shear only. Where the distance between the bearings is greater, and where there is a space between the hub of the sheave and the bearing, Fig. 182, the rule is as follows, and is essentially the same as for a drum shaft under similar conditions.

*Rule for Shaft with Bending Stress at Center.* Multiply $R$, in *pounds*, the total resultant reaction at the shaft center — the same as the total load on the shaft — by $f$ the factor of safety desired, and by $c$, the distance between bearings in inches; then divide the product by 14,400; and the cube root of the quotient is $d$, the required

diameter of shaft in inches. For these conditions the equation is $d = \sqrt[3]{\frac{Rfc}{14{,}400}}$.

*Illustrative Example.* Taking the live load of 3000 pounds again for an example, let the sheave be a top one and set centrally between two beams 12 inches apart. It is desired to find the proper diameter of shaft.

Here the sheave is at the top of the run, as shown in the diagram, Fig. 183, while the engine rests below on a foundation or is hung to the ceiling of a story below. The cables lead up from the driving drum to and over the sheave ($A$) and down to the cage or platform below it. The car, for the sake of illustration, will be considered as not counterweighted from the cage and as weighing 1200 pounds. This load is hanging to the end of the cable which is marked "to cage". The cable is led up to and over the sheave and is secured below, and the whole is at rest. What is the load on the sheave?

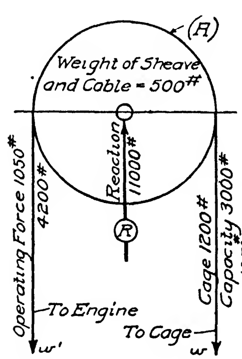


Fig 183. Loads at Top Sheave

While at rest, each half of the cable has the same stress on it, namely, 4200 pounds, therefore the load on the sheave when standing is 8400 pounds. If the cable is attached to the drum below and the latter is made to revolve, the drum has to exert a certain amount of power to move the load — the inertia of the engine balancing that of the load when at rest. Of course the speed with which the load is lifted is what varies the additional force required to overcome inertia, head resistance, and friction, but, in order to avoid prolonging this explanation, it is assumed in this case to be 1050 pounds — 25 per cent of the gross load being lifted. It should be noted that the loading conditions at a driving drum — not shown here — where torsional resistance counteracts a preponderance of load in the cable on one side, are different from those at a sheave pulley ($A$) where, disregarding frictional and bending resistance at the sheave, the aggregate tension in the cable is the same on both sides of the wheel.

Adding the loads on the sheave, the stress $w$ on the cable leading from the cage equals 5250 pounds, and the stress $w'$ — for our

consideration, the same — equals 5250 pounds on the cable leading to the engine, making the total load on the sheave 10,500 pounds, so that, including 500 pounds for weight of sheave and cable, the total reaction $R$ at the shaft journal is 11,000 pounds. A factor of safety of 5 will be found sufficient in this case. Then 11,000 times 5 times 12 divided by 14,400 equals 45.83, and the cube root of 45.83 is 3.57. So the shaft should be $3\frac{5}{8}$ inches in diameter in the body up to the boxes on either side.

The journals may be obtained by the rule for computing the size of gudgeons — thus $d=\sqrt{\frac{5\frac{1}{2}}{.785}}$ — which will make them 2.64 inches in diameter; in practice this would be called $2\frac{15}{16}$ or 3 inches.

**Shaft Carrying Fixed Sheave.** When the sheave is fixed to the shaft and revolves with it but is not in the center — the position for greatest bending stress — between the bearings, a supplementary rule is used in connection with that for the shaft in which bending is considered. It is as follows, referring to Fig. 182.

*Rule for Shaft Eccentrically Loaded.* Multiply the distance $l$ from the center of the sheave to one bearing by the distance $l'$ from the same point to the other bearing; and, after multiplying by 4, divide by the total distance $c$ between the two bearings; which results in the quotient $c''$. Use this consequent center distance, $c''=\frac{4ll'}{c}$, as the distance between the two bearings, and proceed as by the previous rule for the shaft with bending stress; the equation then being $d=\sqrt[3]{\frac{Rfc''}{14{,}400}}$.

*Illustrative Example.* Say the sheave is 20 inches from one bearing and 10 inches from the other bearing, the total distance between bearings being 30 inches. Then 10 times 20 equals 200, which multiplied by 4 and divided by 30 equals 26.67 inches. Take this as the distance between bearings for use in the rule for the shaft under bending stress.

**Loading Effects with Deflecting Sheaves.** When a cable carrying a load is deflected by a fixed sheave, or by an idler on a fixed shaft, the stress on the sheave and its shaft taken in the direction of the main load may be from all to a fraction of the total load on the adjacent parts of the cable, according to the amount of deflection.

Different cases of this variation in loading are illustrated in Fig. 184.

On sheaves, the loads, as $w$ and $w'$, or $w'$ and $w''$, on the cable at the opposite points where it leaves the sheave may be considered as practically equal — the relatively slight differences being due to friction and bending of the cable — and the load $w'$ in the cable between two adjacent sheaves is, of course, the same at both tangent points, so that in connection with idlers we will take the cable ten-

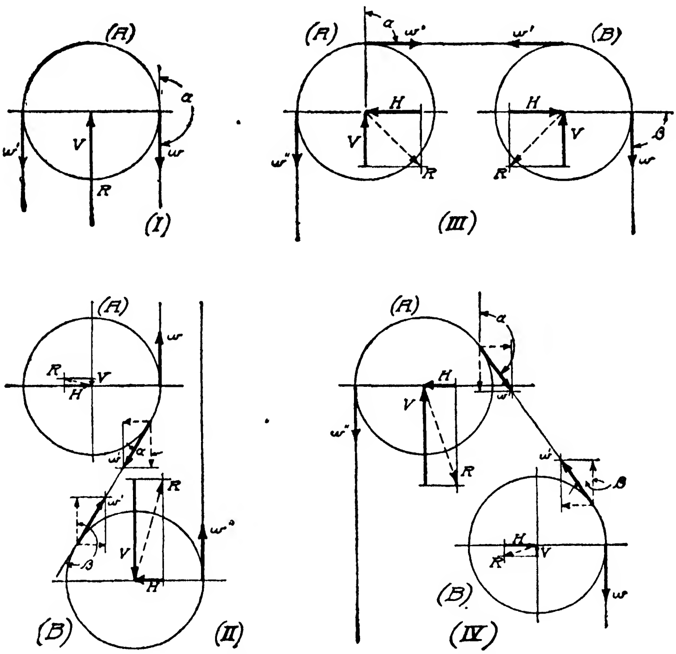



Fig. 184. Diagram of Sheave Loading Conditions

sions $w$, $w'$, and $w''$ as equal. In connection with driving drums, however, the inequality of the tensions in the same cable at the opposite points where it leaves the drum is such — due to torsional driving resistance, as already mentioned — that the difference must be taken into account if the equations involving $w$, $w'$, and $w''$ are used in figuring drum-shaft loading.

*Full Load on Sheave.* In case (*I*), Fig. 184, the conditions are the same as in Fig. 183, where the cable is deflected so far that the full load comes entirely on one sheave (*A*) carrying the cable between

the car and the engine. When, as in this instance, there is no horizontal reaction, or $H_{(A)}=o$, the resultant load $R_{(A)}$, exclusive of the weight of the sheave and cables, equals the vertical reaction, which is $V_{(A)}=w+w'$. In these relations $w$ is the stress in the cable leading from the sheave to the cage on one side, and $w'$ that leading toward the engine on the other side of the sheave.

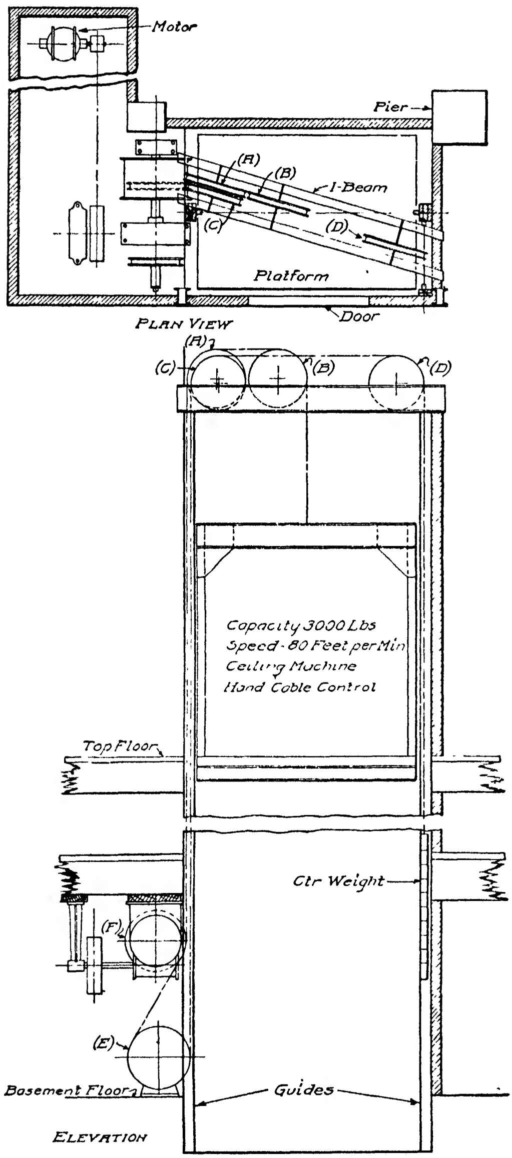



Fig. 185. Single-Belted Electric Freight Elevator

This loading, inverted, is likewise practically what occurs on the sheave $(E)$ at the bottom of the shaft below the belted electric machine of the installation shown in Fig. 185, although the exact proportions are discussed in connection with $(B)$ in case $(II)$.

*Cable Slightly Deflected.* As shown in case $(II)$, Fig. 184, when the cable in passing over the sheave $(A)$ is only slightly deflected from the direction of main load, the vertical reaction corresponding to the load in that direction on the sheave shaft is small and is less than the horizontal reaction at the same point. The

stress on the sheave and shaft from the deflected cable varies according to the angle by which the cable is deflected from a straight line between the driving drum and the load, and may be found by the simple resolution of forces in the required direction.

Making use of the simple trigonometric relations as shown graphically to scale in the force triangles in the diagram, where $\alpha$ is the angle of deflection, the vertical reaction at the sheave $(A)$ — exclusive of the weight of the sheave and cable, as before — is $V_{(A)} = w - w' \cos \alpha$; and, as shown by the direction of the resultant $R_{(A)}$ of the opposing forces acting at the sheave journal, there is a horizontal reaction $H_{(A)} = w' \sin \alpha$. The value of the resultant reaction $R_{(A)}$, or total bending load — as in any case where the rectangular components such as $V_{(A)}$ and $H_{(A)}$ are given — may be found from the relation $R_{(A)} = \sqrt{V_{(A)}^2 + H_{(A)}^2}$.

At the second wheel $(B)$ in diagram $(II)$ the vertical reaction is $V_{(B)} = w'' + w' \cos \alpha$, and the horizontal reaction is $H_{(B)} = w' \sin \alpha$. It should be noted in all these arrangements, where the cable leading to the cage parallels that leading to the engine, that the trigonometric values of $\beta$ the angle of deflection of the cable about the second wheel may be interchanged with those of $\alpha$ the previous angle of deflection, with proper allowance for negative and positive directions. Here the wheel $(B)$ takes practically the whole load carried by the two parts of the cable, as in case $(I)$ at $(A)$ — the exact amount, which is slightly less, being equal to $R_{(B)}$, the resultant of the horizontal and vertical components of the forces acting at this point.

If the angle $\alpha$ is zero — the cosine of 0 degrees being 1.0, and the sine 0, from any table of natural trigonometric functions — the equations at $(B)$ reduce to the same form as for case $(I)$. In fact, it may be further noted in all these relations that the general equations given for the reactions are interchangeable for the several cases defined, with proper substitutions of values.

The conditions shown in diagram $(II)$ exist in such installations as that shown in Fig. 186 — the same as in Fig. 150, Part III of Elevators — the car being electrically operated from the basement floor. In this instance, however, the car is counterpoised from both the cage and the drum separately, so the shaft of sheave $(F)$, Fig. 186, has to carry only a relatively small load, although still in the above pro-

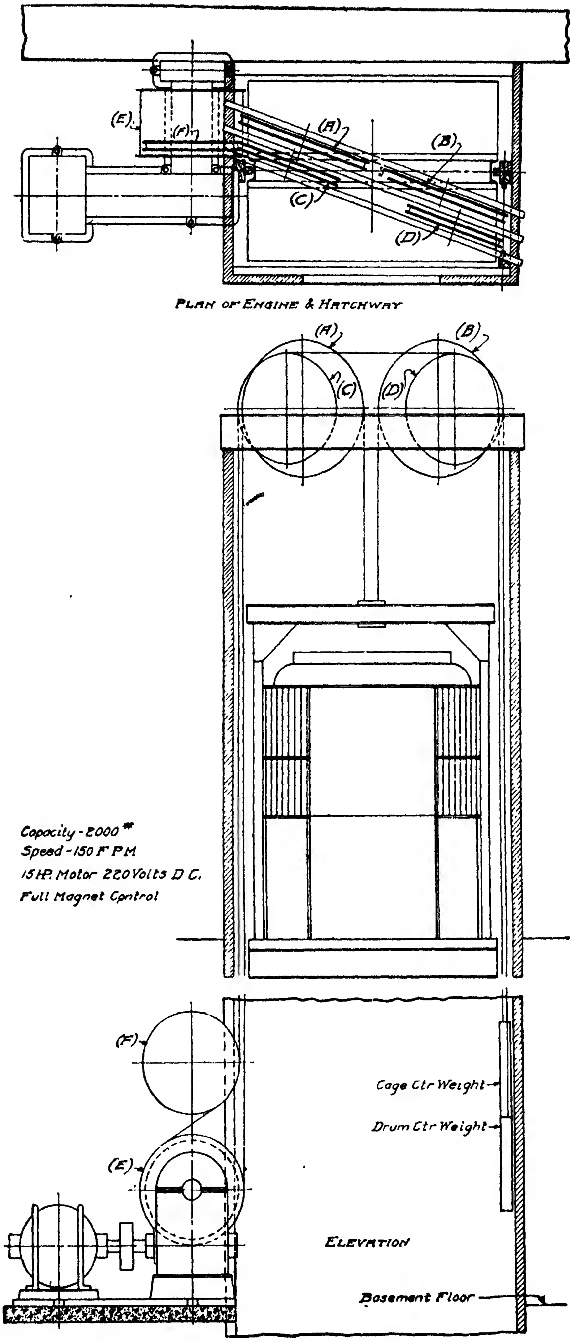



Fig. 186. Direct Connected Electric Passenger Elevator

portions, while drum ($E$) takes the larger part, as is the case with the sheave ($E$) in Fig. 185.

*Two Opposite Sheaves for Same Cable.* When two sheaves are used opposite each other to lead the cables from the engine to the cage, as shown in Fig. 185 at ($A$) and ($B$), the vertical load on each sheave is only half the total load which would be carried by a single sheave when arranged as in case ($I$), Fig. 184.

Diagram ($III$), Fig. 184, illustrates the general application for these conditions. At ($A$) the vertical reaction is $V_{(A)}=w''$, and the horizontal reaction is $H_{(A)}=w'$. At the second wheel ($B$) the vertical reaction is $V_{(B)}=w$, and the horizontal reaction is $H_{(B)}=w'$. This also illustrates the use of the formulas in cases ($II$) and ($IV$) when $\alpha$ equals 90 degrees — the cosine of 90 degrees being zero, and the sine 1.0. The resultants $R_{(A)}$ and $R_{(B)}$ — here, as elsewhere — indicate, however, the combined effects of the vertical and horizontal loads to be allowed for in determining the shaft sizes.

*Cable Acutely Deflected.* As shown in diagram ($IV$), Fig. 184, where the cable is deflected more than a right angle, as angle $\alpha$, the sheave ($A$) carrying the heavier part of the load is subjected to a vertical reaction $V_{(A)}=w''+w'\cos\beta$, and to a horizontal reaction $H_{(A)}=w'\sin\beta$; the supplementary angle $\beta$ — 180 degrees minus $\alpha$ — being used for simplicity. At the second wheel ($B$), where a smaller part of the load is carried, the vertical reaction is $V_{(B)}=w-w'\cos\beta$, and the horizontal reaction is $H_{(B)}=w'\sin\beta$.

This loading occurs in such installations as that shown in Figs. 206 and 207, where the practical requirements of the use of traction machines and limited headroom necessitate it. Where the idler sheave takes the principal weight off the driving drum, the load on the idler shaft may be found as described. On the other hand, if the positions are transposed and the load is borne chiefly by the drum, the sheave shaft should be treated as indicated at ($A$) in case ($II$); which illustrates the general applicability of the formulas with suitable substitution of trigonometric values.

*Idler with Lateral Travel.* The foregoing illustrations indicate the allowances in selecting the sizes of shafts for sheaves to be made due to cable deflection. In addition, there must be considered the lateral location of the sheave between bearings, which may vary the bending stress on the shaft. When an idler travels laterally along

a shaft, it should be treated as though located in the center between the bearings. Using the symbols in connection with Fig. 182, where $c$ is the distance between bearings, the simple equation for the shaft diameter is $d = \frac{R\frac{1}{2}c}{3000}$; in which the resultant reaction $R$ should equal the maximum resultant load on the shaft in pounds.

*Two Idlers on Same Shaft.* When there are two top sheaves, both of them idlers, side by side on one shaft, one of which ($A$), Fig. 185, leads from the drum to the cage, and the other ($C$) leads from the drum to the counterweight, the sheaves will have different resultant loads, $R_{(A)}$ and $R_{(C)}$, both of which must be figured separately and added together, and the shaft must be treated as though the total resultant load $R = R_{(A)} + R_{(C)}$ were at a point central between the two sheaves.

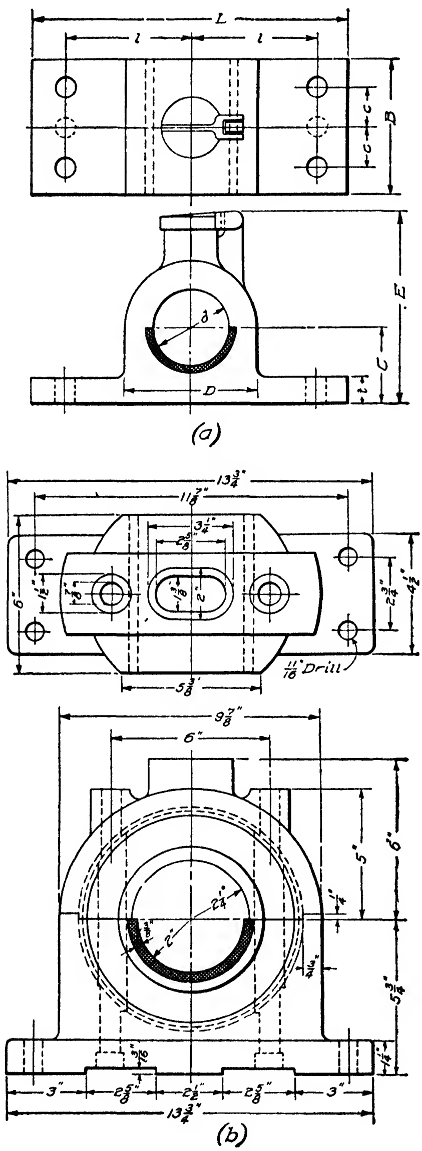


Fig. 187. Bearing Boxes for Sheave Shafts

### Sheave-Pulley Bearings

**Bearing-Box Lining.** The boxes or bearings, Fig. 187, for the shaft journals should be, and usually are, lined with a fair quality of antifriction metal—usually an alloy of lead and antimony, though very rarely with the addition of a small portion of tin.

*Antifriction Metals Used.* It is a misnomer to call this com-

position babbitt metal, for it is not that in any sense of the word — genuine babbitt metal being composed entirely of tin, copper, and antimony in the proportions of 70, 15, and 15, respectively, and the process of melting and blending being somewhat long and tedious. The antifriction metals frequently sold under the name of babbitt metal of various grades, such as X, XX, etc., or by some makers as Nos. 1, 2, 3, 4, etc., are, as described, simply antimonial lead, improved sometimes by the addition of a small per cent of tin or zinc to give it strength to resist pressure.

**Bearing Pressure.** *Intensity.* The reason for going into detail in this matter is to make clear the need for attention to this point of having the lining of the box of a quality that will resist the pressure upon it to the extent of not squeezing out at the ends of the bearing. The United States government specifications for bearings under overhead sheaves for elevators contain the following paragraph.

"Bearings for sheaves shall be lined with antifriction metal; shall be self-aligning; shall be provided with grease cups; and shall be so proportioned that the maximum bearing pressure shall not exceed 350 pounds per square inch."

To give one an idea of the intensity of pressure on a bearing-box lining it is only necessary to divide the load $W$ resting on the bearing by the number of square inches $A$ under pressure, the load being effectively supported on about $\frac{1}{3}$ only of the bearing circumference.

*Illustrative Example.* Take the previously considered case of the sheave with the live-load capacity of 3000 pounds, cage 1200 pounds, etc., Fig. 183. The total working load on the sheave itself — without figuring in its own weight and that of the cables, which is, say, 500 pounds — is 10,500 pounds, but with these added is 11,000. By the rule already given for finding the proper diameter of the journal when the hub is close to it, it should be $2\frac{3}{4}$ inches, or, say, 3 inches in diameter, and 6 inches long. By the $\frac{1}{3}$ rule just given, the load carrying area is 3 times $\pi$ times $\frac{1}{3}$ times 6, which equals 18.84, or about 19 square inches each, and 36 square inches for the two boxes. Dividing the total load or reaction at the bearing, 11,000 pounds, by this effective bearing area shows the working pressure on these bearings to average 305 pounds to the square inch in this case. Now, when it is considered that the linings of the boxes are usually not over $\frac{5}{16}$ inch thick, the necessity for having a fair quality of metal to

line the boxes and for exercising care in this matter can be realized readily.

**Dimensions of Bearing Boxes.** Typical sheave bearing boxes are shown in Fig. 187, (*a*) being the fixed bearing, and (*b*) the self-aligning or ball- and socket-type for top sheaves. The metal of which the boxes are made is always cast iron, and the thickness of metal around the shaft or journal, outside the antifriction lining, is usually $\frac{3}{4}$ inch for journals of 3- or 4-inch diameter, and $\frac{5}{8}$ inch for smaller journals. The foot or base of the box is about the same in thickness, except the part directly below the journal, which is frequently left a little heavier. The bolts used for attaching these boxes to the beams on which they rest are $\frac{5}{8}$ inch in diameter for the sizes mentioned above, which comprise the range of those used for capacities of from 1 to 3 tons. Four are used for each box when located on steel I-beams, but only two each when set on wood beams. Typical proportions of the fixed box are as follows:

**Dimensions of Fixed Bearings**

| Weight (lbs.) | Sizes (in.) | | | | | | | | | Bolts (in.) |
|---|---|---|---|---|---|---|---|---|---|---|
| | $d$ | $C$ | $L$ | $B$ | $t$ | $D$ | $E$ | $c$ | $l$ | |
| 23 | $1\frac{15}{16}$ | $2\frac{7}{8}$ | $12\frac{1}{8}$ | 4 | 1 | $4\frac{1}{8}$ | $6\frac{1}{4}$ | 1 | $4\frac{3}{4}$ | $\frac{5}{8}$ |
| 31 | $2\frac{7}{16}$ | $2\frac{7}{8}$ | $12\frac{1}{8}$ | $5\frac{1}{8}$ | 1 | $4\frac{5}{8}$ | $6\frac{3}{4}$ | $1\frac{1}{2}$ | $4\frac{3}{4}$ | $\frac{5}{8}$ |
| 32 | $2\frac{15}{16}$ | $2\frac{7}{8}$ | $12\frac{1}{8}$ | $5\frac{1}{8}$ | 1 | $5\frac{1}{8}$ | 7 | $1\frac{1}{2}$ | $4\frac{3}{4}$ | $\frac{5}{8}$ |
| 33 | $3\frac{7}{16}$ | $2\frac{7}{8}$ | $12\frac{1}{8}$ | $5\frac{1}{8}$ | 1 | $5\frac{5}{8}$ | $7\frac{3}{8}$ | $1\frac{1}{2}$ | $4\frac{3}{4}$ | $\frac{5}{8}$ |

*Illustrative Example.* The dimensions in connection with the preceding case may now be checked for strength. Refer to the Fig. 185, where two sheaves ($A$) and ($B$) are used for leading the cables from the engine to the cage, or ($C$) and ($D$) to the counterpoise weights, as the case may be. Let it be assumed that the working load or tension $w$ in the cable is 5250 pounds, which is half the total vertical reaction or load to be carried by the two sheaves. The actual pull between the two sheaves, then — that is, the force tending to draw them toward each other — is, as has been already shown, just half of that vertical load on the sheaves, or 5250 pounds.

The sides of the box against which the stress comes in a 3-inch journal have a $\frac{3}{4}$-inch thickness of iron; the width of the foot of the box is 5 inches, and, although the length of the journal was taken as

6 inches, to be sure that the dimensions are safe, let it be assumed that the length of the side of the box is the same as that of the foot, thus ignoring the slight additional strength the $\frac{1}{2}$-inch projection on each side would give it.

In the previous pages, in discussing the transverse stress on the arms of gears, the coefficient given for cast iron was 5544 pounds per square inch. The length of the side of the box is 5 inches and its thickness $\frac{3}{4}$ inch, making the minimum area 3.75 square inches on one side of the 3-inch journal, but as there are two boxes the aggregate wall section for $\frac{1}{2}$ of both boxes is 7.5 square inches. The combined strength of the two boxes at this minimum section therefore would be $5544 \times 7.5$, or 41,580 pounds, leaving a large margin of safety of almost 8 to 1 over the actual pull. However, this is not the total strength of the box in reality, as there are the additional factors of the fillet where it joins the foot and the tensile strength of the opposite sides of the boxes and that derived from the cylindrical shape of the box, which have been ignored, but enough has been seen to prove that the box is amply strong for the work.

**Method of Fastening Sheave Bearing Boxes.** Now as to the bolts; there are four to each box, or eight on a side, that is, eight


Fig. 188. Method of Fastening Bearing Boxes on Wood Beam

to each sheave. The stress on the bolts in this case is purely shearing. Of course, if they are tightened up well, the friction of the bottom of the box on the beam will hold it sufficiently to prevent any shearing strain on the bolts, but bolts frequently loosen from various causes, vibration being the principal one.

*Resistance to Side Stress.* Consider the bolts as being not tight, or only tight enough to keep the box in its place on the top of the beam, so that the force tending to pull the sheaves together has full play. The ultimate shearing stress for mild steel, such as bolts, rivets, etc., are made of, is considered by authorities on the subject to be 7500 pounds per square inch, so a $\frac{5}{8}$-inch bolt having an area of .3067 square inches in shear will withstand 2300 pounds of shearing

stress. There are eight such bolts which, together, are capable of withstanding 18,400 pounds tension between sheaves, or $3\frac{1}{2}$ times the actual force of 5250 pounds in the case considered tending to pull the boxes away from their position.

*Use of Spacer on Wood Beam.* In case the boxes are set on wood beams, this stress is taken care of by spiking a 1- by 5-inch board long enough to fit between the boxes on top of the beam, as shown in Fig. 188. The lag screws then have simply to hold the boxes in place sidewise.

## WIRE CABLES

**Use of Materials.** When employed for hoisting purposes, wire ropes, or cables as they are usually termed, are made of the very best quality of soft iron. It is true that some hoisting ropes are made of steel and that when new they have a greater tensile strength than iron, but they are not so durable, for the reason that they crack sooner.

One of the chief requisites for a durable wire hoisting rope is that it shall be capable of being wound and unwound on and off a drum or spool and roven over grooved wheels or sheaves innumerable times without showing deterioration. It is not the initial tensile strength of a cable that makes it most valuable as a medium for transmitting the power of the engine to the car or counterpoise weights, but rather its lasting qualities under the conditions connected with its work. Hence, ductility is a very important feature, and nothing possesses this desirable quality to a greater degree than does Swedish or charcoal iron. Of course, for special purposes where the duties imposed are more in the line of a direct and steady pull, and where there is little or no bending and unbending of the rope, steel cables will stand a heavier pull than iron. But the winding of a rope around a drum or the leading of it over or around a sheave means the constant alternate bending and straightening out of the cable, and this results eventually in the cracking of the wires of which the cable is composed. Steel, no matter how mild, and no matter how small the percentage of carbon it contains, has a much stronger tendency to return to the crystalline form of structure than iron, and in this condition it is very unreliable.

*Character of Component Wires in Cables.* Perhaps it should be explained here that in the process of the manufacture of iron wire its very method of production — from the time it issues as a bloom from

the puddling furnace and is passed through the rolls down to its being drawn through dies of less and less diameter until its final completion as a wire ready to be made up into cable — all has a tendency to give it a fibrous structure. The absence of carbon from its composition is an important feature in its retaining that structure for a long time under severe service.

Without going into a long discussion of the subject, it may be briefly stated that steel is fundamentally a combination of iron and carbon in certain proportions. In certain kinds, such as in the finer and special grades of tool steel, other ingredients are present, but these are not under consideration here, being wholly unfit for the purpose of hoisting. Reference is made to the Bessemer and open-hearth steels, wherein the proportion of carbon is extremely small, being frequently as low as $\frac{1}{10}$ of 1 per cent, but even this small quantity of carbon is sufficient to start the crystalline formation under the conditions previously described. What probably adds to the tendency is the fact that in the process of manufacture the steels are not subjected to the repeated rolling, welding, and re-rolling that iron is, for this is a part of the process of refining iron which is unnecessary in the case of steel.

*Superiority of Iron for Cables.* It can be set down as an axiom that a good Swedish- or charcoal-iron rope will far outlast one of the same diameter of the mildest steel. Of course the word steel sounds strong, and in writing their specifications many people call for steel cables on their elevators, because of a lack of knowledge of the nature of the metal and because of the popular belief in its strength.

**Construction of Cables.** *Laying Up.* All elevator hoisting cables are made up of 114 wires. These are divided into 6 strands, as they are called, of 19 wires to each strand. The 19 wires composing each strand are separately twisted together as a means of keeping them united as a strand, and the 6 strands are afterward twisted together around a center of hemp cord to form the rope or cable. This operation is technically called *laying up* the rope. The operations described above, while done separately, are performed almost simultaneously and by the same machine, that of twisting the strands occurring only slightly in advance of that which unites them all into one cable.

TABLE I

**Working Capacity of 19-Wire Strand Iron Hoisting Cables**

| CABLE THICKNESS | DRUM OR SHEAVE MINIMUM DIAMETER | MAXIMUM WORKING LOADS $W$ | |
|---|---|---|---|
| $t$ (in.) | $D$ (in.) | Freight (lb.) | Passenger (lb.) |
| $\frac{1}{2}$ | 20 | 1500 | 1100 |
| $\frac{9}{16}$ | 22 | 1800 | 1350 |
| $\frac{5}{8}$ | 24 | 2250 | 1700 |
| $\frac{3}{4}$ | 30 | 3000 | 2500 |
| $\frac{7}{8}$ | 36 | 4000 | 3200 |
| 1 | 40 | 5000 | 4000 |

*Use of Hemp Center.* This twisting of the wires around one another, and afterward around the *heart,* as the hemp center is called, has a twofold object: (1) the preservation of the rope intact during service; and (2) providing a soft cushion on which the strands may rub when the rope bends on going around a sheave or drum. When the rope is bent the position of each strand changes slightly, those strands which are on the outside being bent on a circle of a larger radius than those on the inner side next the groove of the sheave or drum, and when the cable leaves the drum or sheave this position is changed back again as the cable assumes a straight line. With a load on the cable, the rubbing of the strands together would be very severe and would soon result in the wearing of the wires, were it not for the soft cushion of hemp in the center of the rope.

**Weakening Effect of Twisting Wires.** While the twisting of the wires which comprise the strands is essential to the unity of the rope, the wires are not capable of sustaining quite as heavy a load when in this form as they would be if allowed to hang perfectly straight, the serpentine form assumed in the process of laying up being an element of weakness.

*Illustrative Example.* To illustrate, take for example a rope $\frac{3}{4}$ inch in diameter, the wires of which are of No. 18 gage. The breaking strength of one wire of this gage is 169 pounds. This multiplied by 114 the number of wires comprising the rope equals 19,266 pounds, which minus 17,000 the breaking strength of the rope leaves 2266 pounds as the loss in strength by twisting. This loss in strength represents nearly 12 per cent, and applies equally to all ropes of this description.

**Safe Working Loads for Cables.** Table I shows the proper working loads for various sizes of iron ropes — 6 strands of 19 wires each — in use on elevators, and the appropriate diameters of sheaves or drums for each diameter of rope.

*Rule.* It is good practice to allow a working load of $\frac{1}{6}$ the ultimate breaking strength $T$ of the rope — $W=\frac{1}{6}T$ — for freight elevators, and $\frac{1}{8}$ the ultimate strength for passenger elevators — $W=\frac{1}{8}T$. The minimum diameter $D$ of the sheave or drum, as previously stated, is 40 times the thickness $t$ of the rope.

The reason that a smaller load is recommended for passenger elevators is partly to give additional safety where human life is concerned, and partly because passenger elevators usually run faster than freight elevators and the wear is consequently greater. An additional element of safety is also introduced in using two ropes or more, especially when passengers are carried.

**Lubrication of Wire Rope.** Wire ropes should be oiled occasionally, raw linseed oil being the best lubricant for this purpose because it dries gradually and does not run off like machine oils. It can be best applied by means of a brush. A small quantity of good lubricating plumbago mixed with it and kept constantly stirred during the application is an improvement.

**Life of Cables.** Under favorable conditions and with good care, the time a rope will run if in continuous use is about 3 years, ordinarily. Many run a longer time, but under no conditions should a hoisting rope be used for more than 5 years, even if at the end of that time it shows no signs of deterioration. Changes take place in its atomic structure; the constant bending and straightening in passing around the sheaves and drums and the constant strain of the loads it is subjected to all tend to effect changes in its conformation which are not visible. The structure gradually changes from fibrous to crystalline, in which condition its cohesive strength is not so great and it is liable to give out under sudden or severe strain, even though the strain is only momentary. For this reason no dependence should be placed on a rope after it has seen service for a period of from 4 to 5 years.

*Examinations.* During the life of the rope frequent examinations should be made with a view of ascertaining whether it shows signs of wear or flaws in the wires of which it is composed. These

examinations should be made at least every 90 days. The car and weights should be blocked and all strain taken off the ropes, and they should be examined then from end to end at intervals of 20 or 30 feet, special care being given to those portions of the rope which pass over the sheaves and drum. The rope should be slack when examined and should be slightly untwisted so that the interior can be seen, in order to observe if wear or cracking of wires is taking place inside, where it would not be discernible by superficial observation. The wires usually crack on the outside of the rope, but not always. When these cracks occur on the outside, they are readily seen, or may be felt by passing the hand over the rope, but when they are on the inside, nothing but an examination such as described will disclose them.

### Cable Fastenings

**Characteristic Types.** Cable fastenings should be at least 20 per cent stronger than the cable itself. There are many ways of attaching the cables, all possessing more or less merit. The two most popular methods are (1) the socket, and (2) the clamp. The clamp is almost invariably used for attaching or securing the ends of cables inside the drums, and the socket for attachment to cages and counterpoise weights.

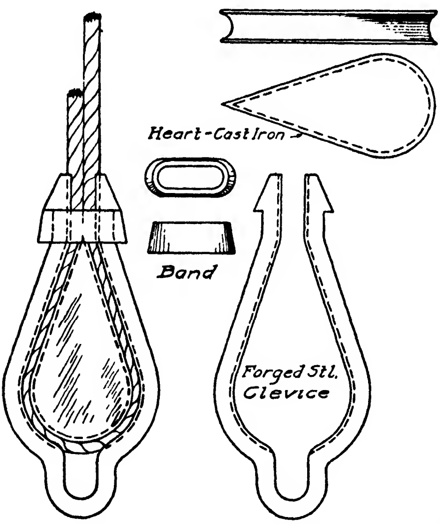


Fig. 189. Special Form of Shackle

**Forms of Cable Sockets.** Sockets vary in shape, Figs. 189 to 192. Some are made conical in form, and are simply slipped into recesses in plates attached to the under side of car beams, Fig. 234. Some, in the form of shackles, are made with lugs for attachment to lifting straps or to eyebolts. Some are made to receive two or more cables, and others for only one. All are usually made of steel or of bronze.

**Strength of Socket Materials.** The following is the basis used for the strength of sockets. The tensile strength of good wrought

iron is from 50,000 to 60,000 pounds per square inch of sectional area; that of open-hearth basic steel 60,000 to 68,000; that of phosphor bronze or aluminum bronze is still greater, being as high as 70,000 to 90,000 pounds. These are the materials of which the sockets are usually made.

When the socket is of wrought iron, a good clean iron of fair quality is essential in order to be sure of a safe weld, for the socket has to be worked out on the anvil from the flat bar and afterward bent around a mandrel and welded together. If a good quality of

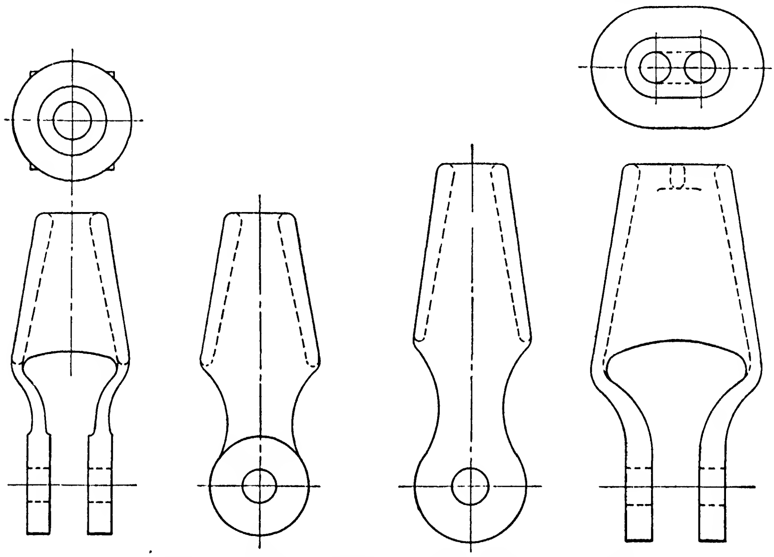


Fig. 190. Standard Shackles for Single and Double Cables

iron is not used, the weld will not be dependable, and for this very reason Bessemer steel is out of the question for making these sockets.

**Formation of Cable Shackles.** *Forgings.* The bar is first formed as shown at ($A$) in Fig. 191, two pieces of which are welded together to form a single piece, as shown at ($B$). The part $MM$ is then bent around a cone-shaped former or mandrel, and the edges $NN$ are welded together, giving the shape shown at ($C$). Following that, the lugs $PP$ are separately formed to shape, bent into position, and drilled for the bolt as shown at ($D$). The finished clevis, or shackle as it is then called, is then ready for use.

*Castings.* When the shackles are made of steel or bronze they are cast from patterns, and being castings and therefore not so dense

in texture as a forged shackle, and also being liable to have other imperfections common to all castings, the metal is left heavier all through them.

**Strength of Socket.** *Bowl of Forging.* The safe limit allowable for tensile stress is 12,000 pounds to the square inch for wrought iron, leaving a proper margin for safety. Consider the load acting as a bursting stress on the bowl of a shackle such as shown in Fig. 191. If this part of the socket is 3 inches long and an average of $\frac{1}{4}$ inch

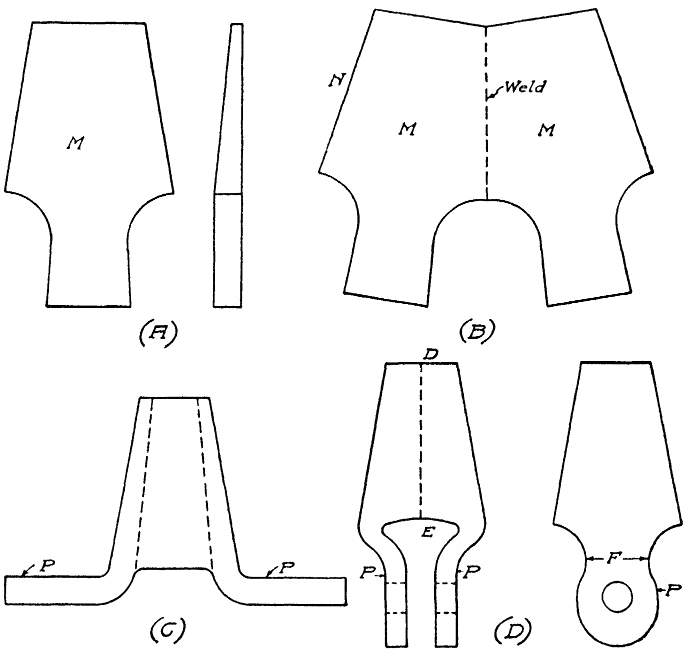



Fig. 191. Method of Forming Forged Socket

thick, the resisting area or the effective section through the bowl from point to base — from $D$ to $E$, in ($D$) — is twice 3 times $\frac{1}{4}$, which is equal to 1.5 square inches. Suppose the live load to be 3000 pounds and the car to weigh 1600 pounds, making a total of 4600 pounds, which we will consider as uniformly distributed as a bursting stress on the bowl. The socket has a safe tensile strength to resist bursting of 1.5 times 12,000 pounds, or 18,000 pounds. This shows the shackle to be amply strong, leaving a large margin for defects in workmanship, should any exist. But this is only for the cone or socket.

*Lugs of Forging.* The lugs, which are two in number, are usually made 2 inches wide by $\frac{3}{8}$ inch thick at the part marked $F$, Fig. 191, ($D$). Hence their combined cross-sectional area here is equal to twice $\frac{3}{8}$ times 2, or $1\frac{1}{2}$ square inches, which it will be readily seen by comparison with the above is strong enough. That portion forming the eye through which the bolt passes is made with the same net sectional area as that at $F$, and is usually 1 by $\frac{3}{8}$ inches in the half

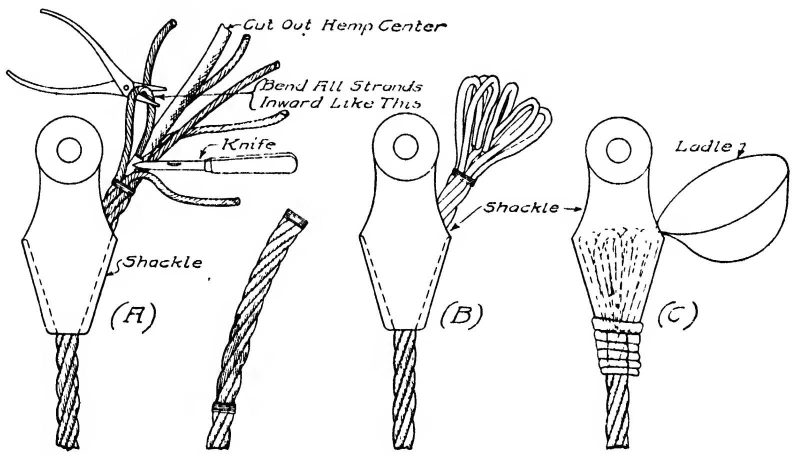



Fig. 192 Standard Method of Shackling Cable

section outside the hole, except directly below the hole where it is usually wider by $\frac{1}{4}$ inch.

*Castings.* In the case of the castings, 5000 pounds to 6000 pounds per square inch is allowed as the safe limit and the patterns are left correspondingly heavier.

**Method of Shackling Cable.** The method of fastening the end of the cable into the socket is shown in Fig. 192, the directions for which are as follows.

Put a seizing of twine on the extreme end of the cable and another at some distance back from the end, the proper position for the latter being twice the depth of the bowl of the shackle. Enter the cable as in ($A$), and remove the seizing from the extreme end and untwist the cable back to the second seizing, separating the several strands. Cut out the hemp center close down, then turn each strand separately inward as shown in ($B$). Make each strand form a loop, all the ends meeting in the center down close to where the heart is cut out. Draw the looped end back into the socket or bowl of the shackle, but not too tight — there must be left some interstices for the babbitt to run into, and thus form a lock. Wrap rags around the place where the cable enters the shackle to prevent

the melted metal running out there, then fill the bowl or socket with melted babbitt; see that the metal is hot, so it will not chill too soon and thereby fail to run into every crevice. At the same time care must be taken that the melted metal is not red hot, or it will anneal the wires and destroy their tenacity. The process of drawing the wires through the dies used in their manufacture compresses them to such an extent as to give them a sort of temper which contributes greatly to their strength. When a fire occurs in a building and any cable is exposed to sufficient heat to anneal it, the cable is useless and must be discarded, even though the annealed part is only 1 or 2 inches long.

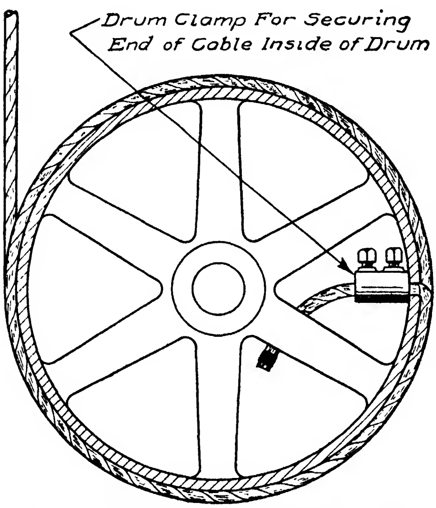



Fig. 193. Method of Clamping Cable on Drum

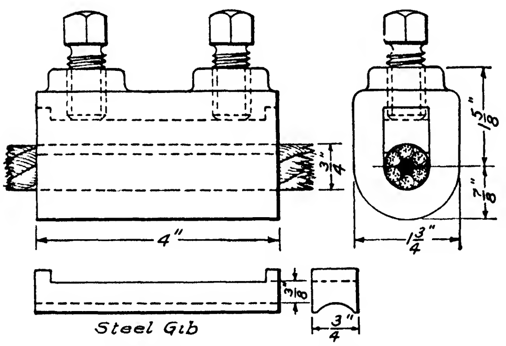



Fig. 194. Detail of Drum Clamp

**Drum Clamps for Cables.** Drum clamps are shown in Figs. 193 and 194, and need no further description. It is well to state, however, that they are not subjected to nearly so much stress as the shackle or the socket which is used with the beam plate, because the cable is always wrapped at least twice around the drum before the end is passed through for the attachment of the drum clamp. The sharp turn which the cable takes in entering the drum, as well as the friction or adhesion to the drum caused by the two or more wraps of cable around it, is almost enough in itself to hold the cable safely, the office of the clamp being rather to hold the end of the cable in place than to take any great stress. Still, these clamps are usually made of steel castings to insure them against being cracked by tightening the set screws too hard, cast steel being proportionately stronger than cast iron.

## SUPPORTING STRUCTURE

### GUIDEWAY CONSTRUCTION

**Speed Factor.** In the earlier days when the elevator was not such a complicated machine as it is at the present time, the speeds were much slower and it was not so essential to have such good guides, guide shoes, and runways as it is today.

#### Wood Guideposts

**Solid Wood Guidepost.** *Construction.* As the timbers were more readily obtained in those days, the guideposts were usually made of one solid piece, the sizes used being either 6 by 6 inches or 6 by 8 inches for the smaller elevators and 8 by 10 inches for the large heavy freights. These timbers were dressed on all four sides, and that face to which the maple guides were to be attached was carefully straightened and trued from end to end.

*Objectionable Features.* It was found, however, that even when the greatest care was taken to select only well seasoned stock, certain changes were inevitable for a number of months after the guideposts were installed. The further drying out of the heart of the lumber caused the posts to crack and twist. Sometimes the fitting of them between the floors was done too tightly, while a settling of the floors in the old buildings frequently would bend the posts. Therefore this form of guidepost gradually was superseded by what is called the sectional or compound post in wood.

**Sectional Wood Guidepost.** *Construction.* By using three or four 2-inch well seasoned planks 6, 8, or 10 inches in width, fastened together face to face after dressing, with the grain of each plank opposed to that of the one to which it is spiked, the planks serve to correct one another, thus forming a better guide. Regardless of how well a piece of lumber has been seasoned, it always has a tendency to warp, shrink, or otherwise change its shape after being placed in a warm, dry building. A man experienced in the handling of lumber can tell at a glance in what direction a piece of lumber is likely to warp, and hence is able to spike several pieces together so that the warping tendencies are counteracted. Figs. 195 and 196 illustrate the methods of putting these planks together to form the guideposts. In selecting the planks and in putting them together, it is generally

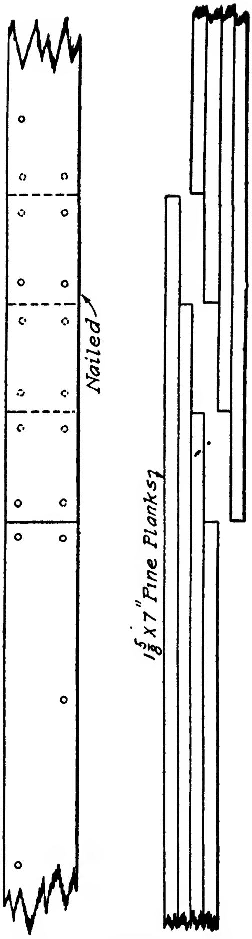


Fig. 195. Wood Guidepost

done in such a way as to form lap or splice joints at the ends, as illustrated.

**Method of Setting Posts.** In installing the posts instead of shouldering them back under the joists and over the floors, as detailed in Fig. 198, (*a*), they are put wholly within the hatchway. By this method the settling of the floors does not affect the posts so badly, and another advantage is that no particular lengths to suit each story are necessary. In spiking the posts together, care must be taken to stagger the spikes and to have them long enough to go through more than two thicknesses of the wood.

*Use of Plumb Lines.* When setting up the posts in the hatchway they are always set to plumb lines dropped from a plank laid temporarily across the center of the hatchway at the top of the building, as shown in Fig. 197. By means of a plumb line dropped from the edge of this plank, the center of the hatchway is first determined by measurements from the trimmers of each floor, after which other plumb lines are dropped $\frac{1}{2}$ inch from the plane of the guideposts and fastened to a plank at the bottom of the shaft. The guideposts are then set parallel to these lines.

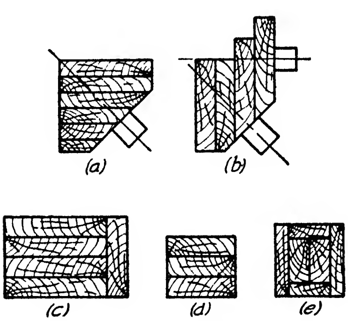


Fig. 196. Sections of Wood Posts

In the better class of work where absolute accuracy is desired, after dropping the plumb line and marking its position below as well as above, it is removed entirely and a fine steel wire is substituted. The use of the wire instead of the linen plumb line is preferable, in that the wire can be drawn much tighter and is not so affected by the weather as a linen or silk cord is. These plumb lines are

run ½ inch from the faces of the guideposts, because if run next to the faces they could easily be bulged out of position. The distance of

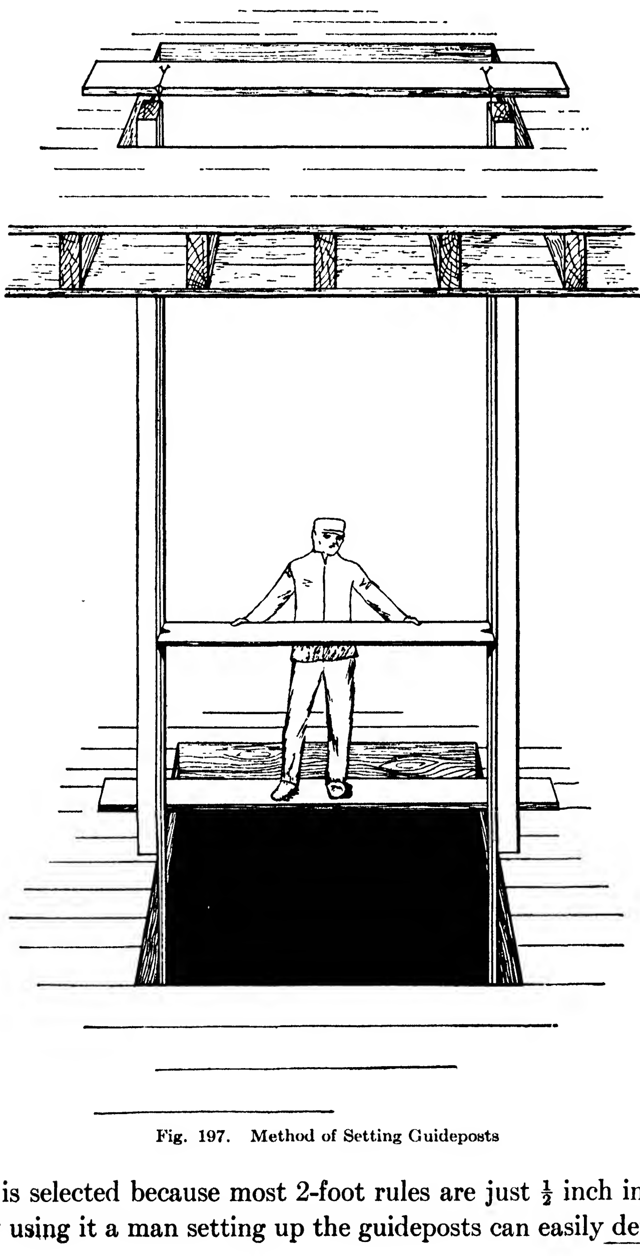


Fig. 197. Method of Setting Guideposts

½ inch is selected because most 2-foot rules are just ½ inch in width, and by using it a man setting up the guideposts can easily determine

when the post is set exactly parallel with the line. Should the post be too close by even a minutely small fraction, the $\frac{1}{2}$-inch rule would not pass between it and the line without disturbing the line, while if the post is too far away light may be seen through the rule clearance.

To set the post plumb, or fairly so the other way, a line is marked the whole length of the post in the center of its face. Upon this face a square is used, and when the heel of the square is set to the line marked on the post with the stock of the square on the face of the post, the blade should just touch the line and no more.

*Use of Gage Board.* In addition to these precautions a gage board is used, as shown also in Fig. 197. This is a piece of dry 1-inch board dressed on both sides, one edge straightened and both ends squared to this straight edge. The length of the gage board must

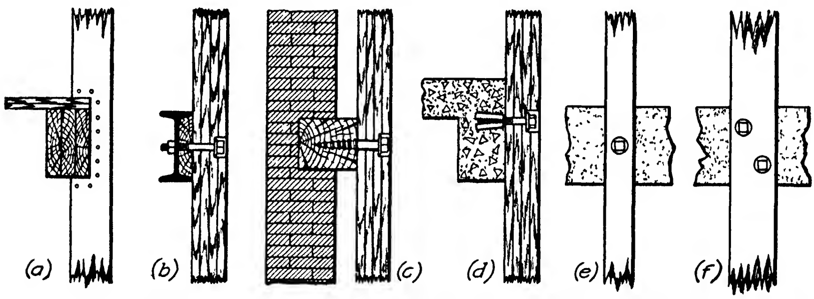



Fig. 198. Methods of Anchoring Wood Guideposts in Hatchways

be exactly the distance apart required between the faces of the guideposts. In order that the using of the gage board shall not interfere with the plumb lines, a V-shaped notch is cut in each end of the board, so that in its application to the guideposts this V straddles the line and does not touch it.

This board is used to detect any tendency of the guideposts to wind or get out of square. Hence it is customarily used after the guideposts in each story are set in place and fastened sufficiently to hold them in position but not so securely as to prevent them from being shifted easily. If it is found that the posts are not exactly parallel to one another, it is an easy matter to remedy it either by shimming, or, if too close together, by removing the post and paring off the back or paring off the hatch trimmer.

**Wood Post Installation in Steel Hatchway.** Where the frame of the hatchway is of steel, only the post can be pared, although shimming is easily done where required. The bolts supporting the posts are put through the I-beams, the heads being countersunk below the face of the post and a nut screwed on the other side of the web of the beam, Fig. 198, (*b*).

**Wood Post Installation in Brick or Concrete Hatchway.** In the case of brick or concrete hatchways, other methods are employed, Fig. 198, (*c*), (*d*), (*e*), and (*f*), depending upon the conditions and surroundings. In case the walls of the brick hatchway are not more than 12 inches thick, or those of the concrete hatch not more than 6 or 8 inches, some builders prefer to drill holes through the wall and to bolt the posts to the wall itself. This is undoubtedly the more substantial method, because, should the lumber in the posts shrink at any time after they are installed, access can be had to the nuts outside the wall to tighten the bolts.

*Use of Shell Bolts.* In many cases, however, what are called shell bolts are used, Fig. 198, (*d*). They are bolts made with long nuts, usually of cast iron, and in two parts. The nut is about $1\frac{3}{4}$ to 2 inches long and is divided longitudinally into two halves. It is made in such a way that when the bolt is screwed into it the end of the bolt shell expands. The hole for the shell or nut is drilled $2\frac{1}{2}$ to 3 inches deep and the shell is slipped in. As the bolt is screwed into the shell it spreads the inner ends of the shell, causing them to grip tightly on the sides of the hole in the wall and to hold firmly. However, it requires some skill and judgment upon the part of the workman making the hole to obtain the proper size and depth. A failure fully to understand the conditions frequently has resulted in unsatisfactory work.

*Bolting.* In bolting the guideposts their shape and structure must be taken into consideration. With a post 6 inches wide, one bolt in the center is usually sufficient, Fig. 198, (*e*). When they are set up between trimmers in an open hatch of course they can be fastened only at their ends; but where they are set up against a brick or concrete wall, the posts must be fastened at the end and also in the middle. In putting fastenings in the middle of the post between the ends, care must always be taken to make the back of the post touch the wall firmly, for, if there is a space between the post and

the wall, the tightening of the bolts will draw the post back firmly against the wall and pull it out of line.

Where the posts are 8 or 10 inches wide it is a better plan to stagger the bolts, Fig. 198, (*f*). This has a tendency to keep a wide post from twisting. In all cases where bolts are used, they must be countersunk to bring the heads below the surface; otherwise they will be in the way of the maple guides and of the guide shoes. In countersinking for the boltheads care must be taken to make the countersunk portion of the hole large enough in diameter to admit a cut washer under the head of the bolt, otherwise it will not have sufficient bearing on the lumber to prevent its being drawn in and thereby allowing the post eventually to become loose.

### Wood Guides

**Car Guides.** *Installation.* After the guideposts have been set up accurately to the lines, the latter are removed and set up again. This time the lines are so set as to be $\frac{1}{2}$ inch from the *side* of the guide strip, usually of maple, which is to be put on. The faces of the guideposts, having been set to a plumb line, are in themselves a sufficient guarantee that the maple guides will be put on straight in that direction. But it is just as important that they be straight sideways — that is, toward the back or front side of the hatchway. Hence the lines are set the second time so that the maple guides may be properly aligned in this direction. These maple guides are usually attached by means of wood screws, as shown in Fig. 199. As a matter of convenience, maple guides are usually made in short lengths of about 4 feet.

*Construction.* After being dressed, squared, and smoothed, a tongue is made at one end of each length and a groove at the other, as detailed in Fig. 199. Several lengths are then fitted together and planed again with the hand plane in order to remove any unevenness at the joints. They should then be given a coat of raw linseed oil so that no changes will take place until they are put into position.

If this precaution of matching the ends is not taken, the screws which hold the guides to the guidepost will not be found sufficient to keep the guides in perfect alignment. The tonguing and grooving of the ends makes it impossible for them to get out of line with one another after being securely fastened in place. However, even after

they are put up in position, it is necessary to examine the joints to discover and remedy any further unevenness that may occur.

*Sizes of Wood Guide Strips.* For ordinary work for cars not exceeding 6 or 7 feet square, the most popular size for the guide strip is $2\frac{1}{4}$ by $2\frac{1}{2}$ inches, although they are often made $2\frac{3}{4}$ by 3 inches. For cars which are to handle very heavy loads, and for those which are of large size or abnormally long, maple guides of this size would not be sufficiently strong to withstand the strain and shock of putting on heavy loads, and for such cases the guide strips are made about $3\frac{1}{4}$ inches square. In the case of excessively large platforms 25 to 30 feet long and 8 or 9 feet wide, Fig. 223, guide strips approaching 4 by 6 inches would be used.

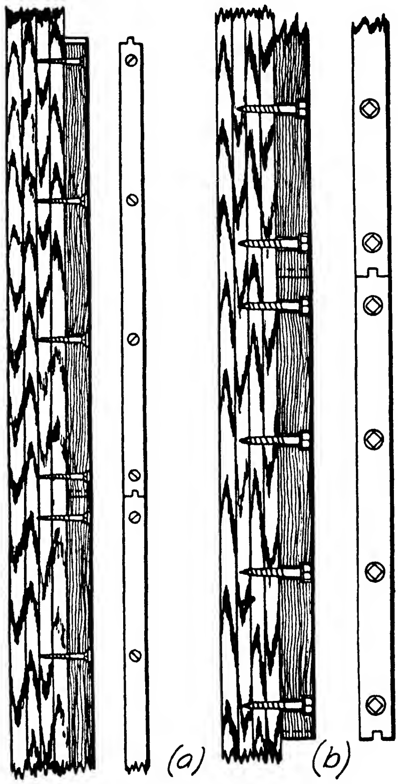


Fig. 199. Methods of Fastening Maple Guides

*Material in Wood Guide Strips.* Maple is used for making guide strips because it is a close-grained wood, does not sliver, presents a hard surface, and has many qualities which fit it peculiarly for this kind of work. Beech or birch can be used, but good hard well-seasoned maple is the best. Although much has been said about the liability of maple to warp, these troubles are experienced only in the getting out of the strips in the shop. After they are once oiled and put in place, they are covered with a coat of grease for the purpose of lubrication before the elevator is started, and after this time the coating of grease which must be constantly maintained protects them from any influence of the weather.

**Counterpoise Guides.** Similar but smaller wood guide strips are used in connection with the counterbalance weights. Those used on ordinary power elevators for the counterpoise weight are similar

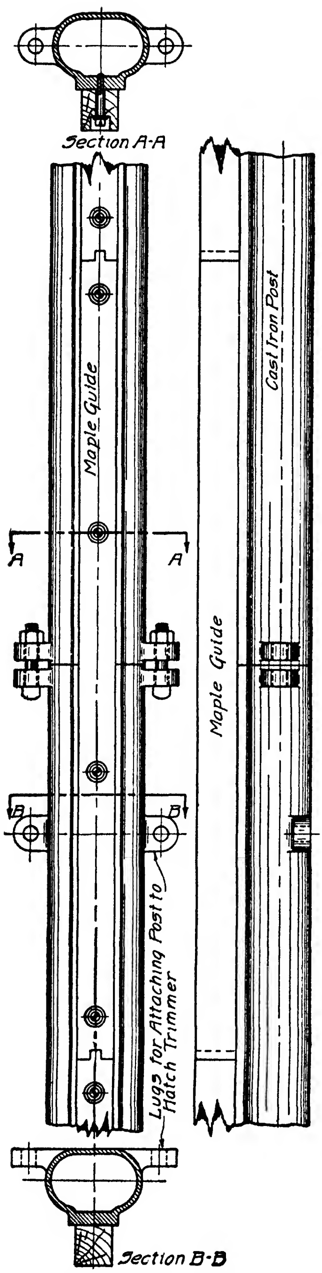



Fig. 200. Iron Post with Maple Guide

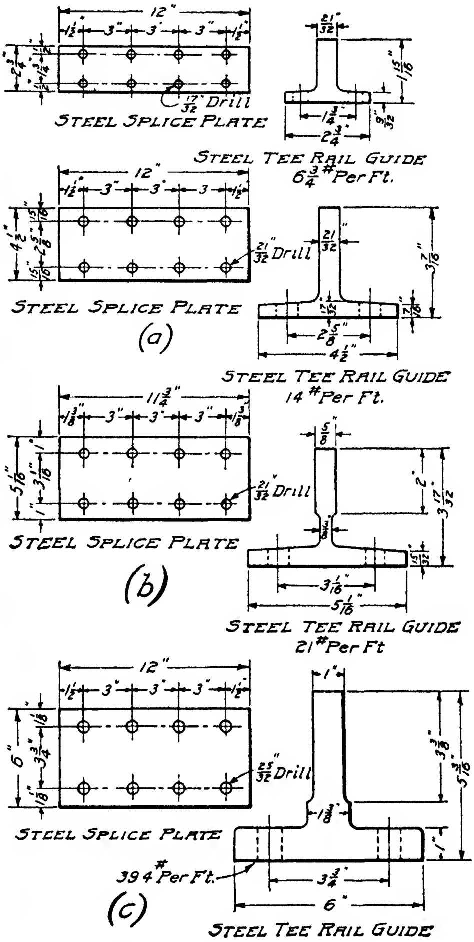



Fig. 201. Details of Steel Guides

in size to what is used for the platforms of hand elevators, that is, $1\frac{3}{4}$ inches square.

*Means of Fastening.* The small sized guides are fastened to the posts by means of No. 18 flathead wood screws $3\frac{1}{2}$ inches in length, countersunk so as to be below the surface, Fig. 199, (*a*). Too much emphasis cannot be placed upon the necessity for having all screw heads

on guide strips sunk well below the surface, for otherwise serious damage may result. For guide strips $2\frac{1}{4}$ by $2\frac{1}{2}$ inches the wood screws used are similar to those for the $1\frac{3}{4}$-inch guide strip, but usually are of about No. 22 size 5 inches long. For maple guides $3\frac{1}{4}$ inches square lag screws are best, each lag screw having a cut washer placed under the head when used, and the strip countersunk to suit, Fig. 199, (*b*).

### Steel Guideway Construction

**Steel Guide and Post Combined.** *Development.* For high-speed elevators, especially in the case of long runs and in those cases where a more smooth and durable runway is desired, guides made of steel have come into use. These guides combine the functions of both

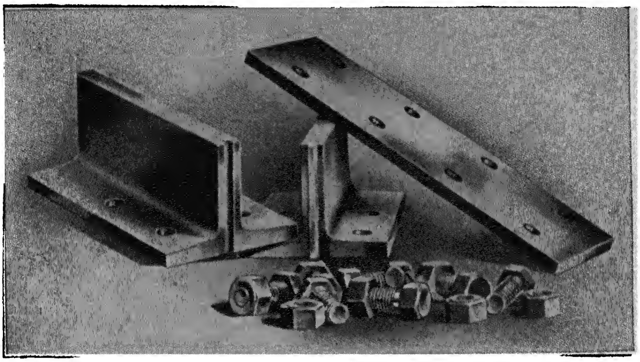


Fig. 202. Parts of Steel Guides Ready for Assembly

guideposts and guides. They were slowly developed, the first metal guideposts being of cast iron and having maple guides, Fig. 200. These passed quickly out of use.

*Construction.* The first steel guides were made by taking ordinary steel T-bars and milling the sides and top of the rib, the guide shoes being made to run on them. Later, special shapes were rolled with a thicker portion for planing or milling. Today they are produced by taking a certain size of steel T and drawing it through dies. This produces a smooth straight surface at a comparatively low cost. The main elevator guides are always of T section, the back being about $4\frac{1}{2}$ inches in width and the web $3\frac{1}{2}$ by $\frac{5}{8}$ inches, and the weight guides made of steel are of similar shape but smaller, being $2\frac{3}{4}$ inches in width and having a $1\frac{15}{16}$-inch rib. Sections for different capacities are detailed in Fig. 201, at (*a*), (*b*), and (*c*).

Joints usually are made in lengths of 14 or 16 feet, or even longer. Owing to the manner in which they are spliced or joined together, they can be set up in a continuous run without regard to the floors

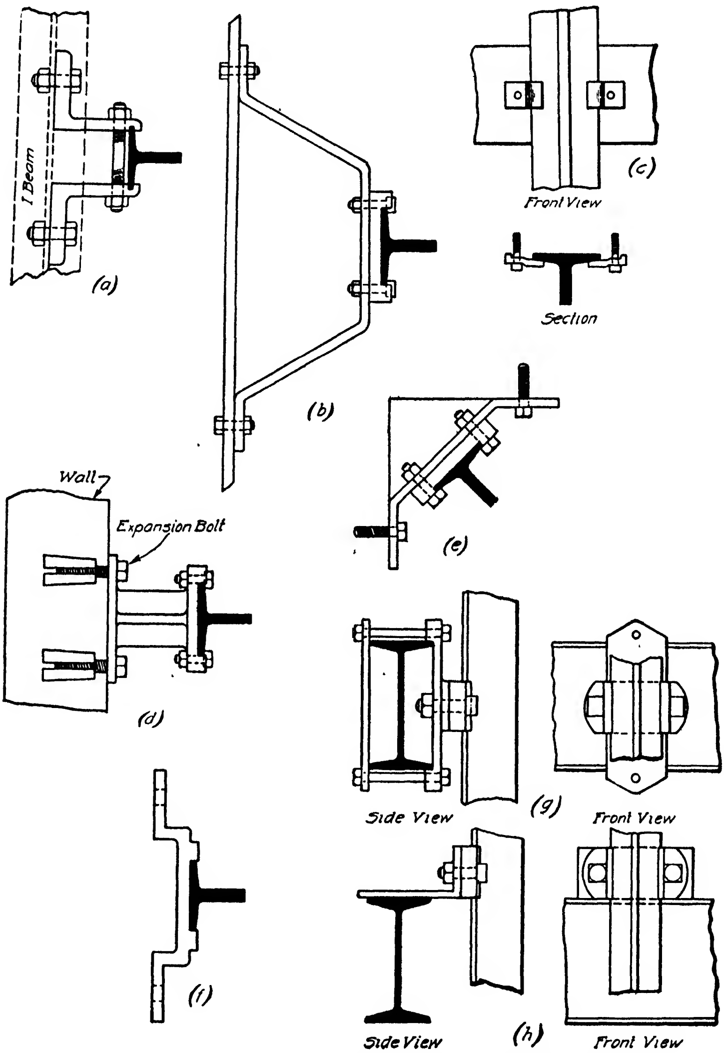



Fig. 203 Methods of Anchoring Steel Main Guides in Hatchways

or landings, the only cutting necessary being on the last length at the top, and hence they form one continuous rail from the bottom to the top of the building. The ends of the guides at the joints are held

in place by tonguing and grooving, and are fastened together by a plate bolted to the back of them, Fig. 202.

*Attachment.* The guideposts are held in position by means of brackets arranged as shown in Figs. 203 and 204, which are made

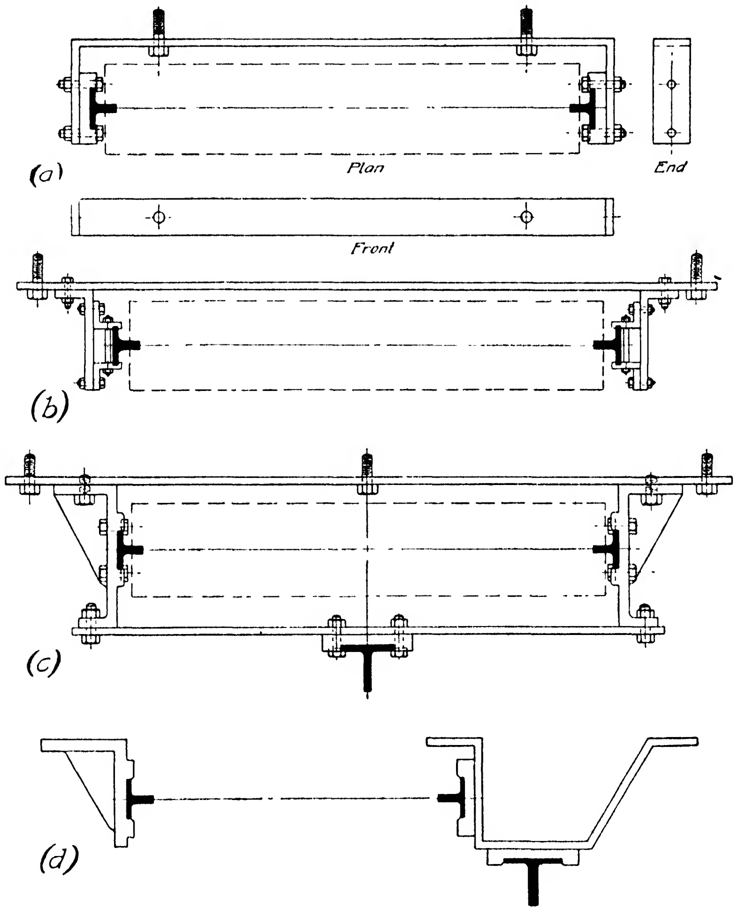



Fig. 204. Methods of Anchoring Steel Counterweight Guides

either of wrought or of cast iron, as the necessities of the case require. These have a shoe in which the guidepost fits, and in which it is held firmly by two bolts. The method of setting them is similar to that used for setting the wood guideposts. A gage board is also used, being fitted with jaws of metal at its ends which fit the guides. These

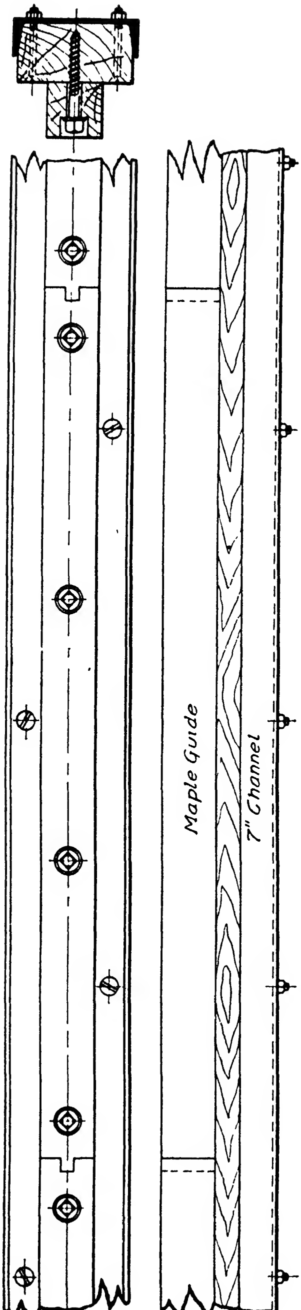



Fig. 205. Channel Guidepost

jaws are removable and can be attached to any gage board. As these guides act also as guideposts, only one setting is required.

**Channel Guidepost.** *Construction.* Before leaving this subject, it is well to mention another type of guidepost which is still used to some extent. It is made of an iron channel, sometimes filled with wood, as shown in Fig. 205. The wood is secured to the channel by means of wood screws or by lags, as the size of guidepost and lumber used for filling seems to require.

*Method of Attaching Guides.* When made in this way, the maple guide is attached by means of wood screws to the wood filling. However, when the channel alone is used without a filling, it is customary to use a somewhat heavier channel, and the maple guide is attached to the back or web of it. When this is done, care should be had to tap or thread the holes in the back of the channel for the bolts which hold the maple guides to it. These bolts should be screwed into the channel and a nut put on afterward to act as a locknut. If this precaution is not taken, the nuts holding the bolts are likely to become loose.

## OVERHEAD BEAMS

**Engine Overhead Arrangement.** A very important feature in the designing of an elevator is the proper selection and arrangement of the beams which are set across the top of the hatchway for the purpose of supporting the sheaves or the

engine from which the car and counterweights are hung. When steam or water under pressure were the mediums for the application of power for the operation of the elevator, the size and shape of the engines used and the difficulty of carrying steam pipes such a distance precluded the possibility of setting the engine overhead, but with the introduction of electricity as a motive power, the compact design of the electric hoisting engine and the facility with which the feed wires could be carried to the top of the runway made this matter comparatively easy.

*Advantages.* The advantages gained by setting the engine overhead are two: (1) the saving of room in the basement of the building formerly occupied by the engine, which in crowded communities where ground space is valuable is no trifling item; and (2) the elimination of many moving parts of the machine, with a consequent reduction in friction and a correspondingly smaller amount of current required for the operation of the plant.

**Comparative Requirements with Different Arrangements.** It might seem to the casual observer that the placing of the engine overhead would necessitate the use of heavier beams and stronger walls to support the additional load. Such is not the case in reality, as is evident from the following considerations.

*Engine in Basement.* In Fig. 185, representing the penthouse and upper works of an ordinary freight elevator of 3000 pounds capacity, the lifting and counterpoise cables at the left run down from the upper sheaves and connect with the winding drum of the engine located in the basement of the building. On the other end of these cables are hung, respectively, the car and the counterpoise weight.

Suppose the car to weigh 1200 pounds as before, and, as the machine is an electric elevator, the counterpoise weight will weigh as much as the car together with from 30 per cent to 40 per cent of the live load to be lifted. The car with its total live load will be 4200 pounds, and it is evident from what has been explained in connection with the section on Sheave-Pulley Shafts, that a similar stress will be exerted on the other part of the cable which leads from the overhead sheaves ($A$) and ($B$) down to the winding drum ($F$), thus making a total weight on the top sheaves and the beams which support them of 8400 pounds. In addition to this there is the counterpoise weight, which for effective and economical service

must equal the weight of the car, 1200 pounds, plus, say, $\frac{1}{3}$ of the live load, or 1000 pounds, a total of 2200 pounds. The same rule will apply in the case of the counterpoise cables as in that of the lifting cables — viz, that the pull on both sides will double it — so that this load on the counterpoise sheaves ($C$) and ($D$) and the beams will be 4400 pounds. These loads combined make a total weight on the beams when the engine is at rest of 12,800 pounds. This is not taking into account the weight of the beams themselves, nor of the sheaves which will weigh, say, about 1200 pounds. So the total static load or weight on the structure, without the beams which for simplicity of illustration may be left unconsidered, is 14,000 pounds in this case.

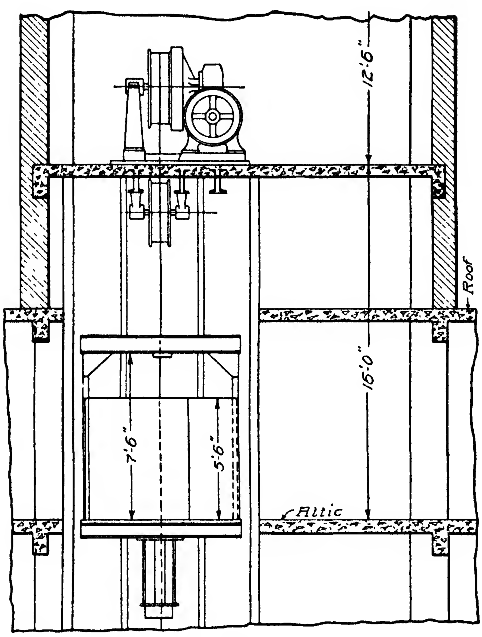
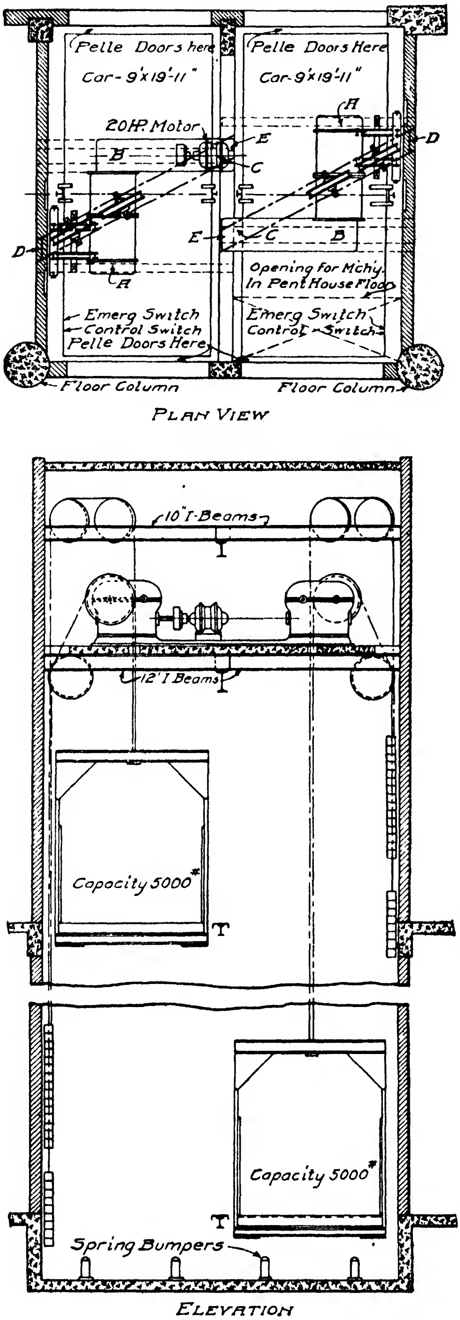



Fig. 206. Traction Machine in Penthouse

*Motor in Penthouse.* An electric engine capable of lifting 3000 pounds or from 80 to 200 feet per minute will weigh, including its motor, about 7200 pounds, and the car with load and the counterpoise weight will be the same as in the preceding case, viz, 6400 pounds, making a total standing load of 13,600 pounds, which is less than in the first arrangement.

**Allowances in Overhead Construction.** Figs. 206, 207, and 254 are diagrams of overhead engine arrangements. These conditions prevail with but slight variation in all cases, whether for a heavy or a light freight elevator, or a passenger elevator of any capacity. It will be seen, therefore, that heavier walls are not needed when the engine is to be set at the top of the run, over the hatchway.

*Dead-Load Factor.* A higher penthouse is required to accommodate the height of the engine, and, as it usually occupies a space fully




KIND OF ELEVATOR Freight
MACHINE Semi-Magnet AA
CAPACITY 5000 LBS. SPEED 90 F P M.
VOLTS 220 PHASE 3 CYCLE 60
MOTOR 20 HP. C.S. #572-720 R P M - 8 Pole Westinghouse
CONTROLLER 20 HP. Reverse Switch-H E & N Co. Magnet Line Switch-C H & Co.
FLOOR STOP
BRAKE Electric-Shoe 18"x7"
SLACK CABLE On The Cross Head
WORM GEAR $1\frac{1}{4}$" P. x 60 T x 24"
WORM $1\frac{1}{4}$" P x $1\frac{1}{2}$" L DRUM 33"x32", 33"x35"
CAR No 3 CAB
WAINSCOTE 2-Sides-72" High Sheet Steel
CONTROL Crank
SAFETY GRIPS Wedge Type
CAR GUIDES 2-$4\frac{1}{2}$"x$3\frac{7}{16}$"x$\frac{21}{32}$" Steel Tees-84 Ft.
CTW GUIDES 2-$3\frac{1}{4}$"x$2\frac{1}{2}$"x$\frac{21}{32}$" " " 84 Ft
CAR CTW 6"x6"x48" —— 5200# 3H-11F
DRUM CTW 6"x6"x48" 3100# 2H-6F
HOIST CABLES Two-$\frac{3}{4}$" Swede 95 Ft
DRUM CTW CABLES Two-$\frac{3}{4}$" Swede 105 Ft
CAR CTW CABLES Two-$\frac{3}{4}$" Swede 105 Ft
GOVERNOR CABLE One-$\frac{1}{2}$" 210 Ft.
OPERATING CABLE $\frac{3}{8}$" Tiller 275 Ft
ANNUNCIATOR 8 Drop (1-2-3-4-5-6-7-8) Single
LIGHT FIXTURES One
EMERGENCY SWITCH in the Car
HATCH LIMITS Two
GATES
ENCLOSURE
SERVICE WIRES BROUGHT TO Within 10 Ft of our Controller By Owners
Cutout Switch & Fuse Blocks By Owners
BEAMS FURNISHED Us
OVERHEAD FLOORS By Others
SUPPORTS By Others

**SPECIAL MATERIAL**

Hinged Screen Canopy Over Car
Guide Oilers
CTW. Guard at Pit 7'-0" High and 6" From Pit Floor
Bumpers at Top of CTW
Bell in Connection With Annunciator

**DOWN LOADS**

| | | |
|---|---|---|
| A 6800 | F | These Are Actual Live Loads and Should Be Doubled for Impact, with 16 000 Pounds Stress for Steel |
| B 7200 | G | |
| C 2400 | H | |
| D 7700 | I | |
| E 3300 | J | |

**The Haughton Elevator & Mach. Co.**
TOLEDO, OHIO, U. S. A.

Fig. 207. Haughton Overhead Double-Drum Machine

as large or larger than the hatchway, a greater floor area in the penthouse will be required. This is all the more necessary for a passenger elevator with magnet control and having two or more speeds, because the controller for such a machine is quite large and occupies about 8 or 9 square feet on the floor, and weighs anywhere from 1000 to 1400 pounds, which

extra weight must not be forgotten in estimating the weight on the walls of the hatchway or the floor of the penthouse.

To the loads estimated as shown in the foregoing must be added also the weights of the beams which are selected to carry the load.

*Impact Factor.* In the selection of the I-beams or channels for this arrangement, another factor enters the case. The loads to be estimated as described above are the dead loads, but when the engine starts in motion, a certain amount of force has to be exerted to overcome the inertia of the car and its load and of the moving parts of the machine. Also, when a stop is made and the current is cut off and the brake is applied to stop the machine, the momentum of the moving parts must be overcome. This produces an additional stress on the beams and on the walls that support them, known among elevator builders as the *impact,* which it really is.

Of course the impact varies with the speed and load in machines of different capacities, and is a somewhat complicated calculation. It usually amounts to from 75 per cent to 90 per cent of the total dead load, so in practice, in order to arrive at the ultimate stress on the beams, it is customary to simply double the dead load, and to select them on this basis.

**Beam Arrangement.** *Sheaves Overhead.* When the overhead work is simply sheaves, the engine being below, two I-beams are used for each set of sheaves, as in Figs. 185 and 186, the boxes in which the sheave shafts revolve being set on these beams and bolted to them.

*Engine Overhead.* When the engine is set over the hatchway, Fig. 151, Elevators, Part III, and Figs. 206, 207, and 254, either I-beams or channels are used, the latter being used in pairs bolted together to form one beam with a space between them to accommodate the holding-down bolts. A plate which spans both channels is used below, and this, while more expensive, is a better way than using I-beams, on account of the greater facility with which the engine can be bolted to them, the beams in all cases being arranged so as to be convenient for bolting the engine down solidly.

**Overhead Floor.** The floor, some kind of which is always used, just above the beams and between them and the engine, may be a grating for the sake of allowing light to pass down the hatchway

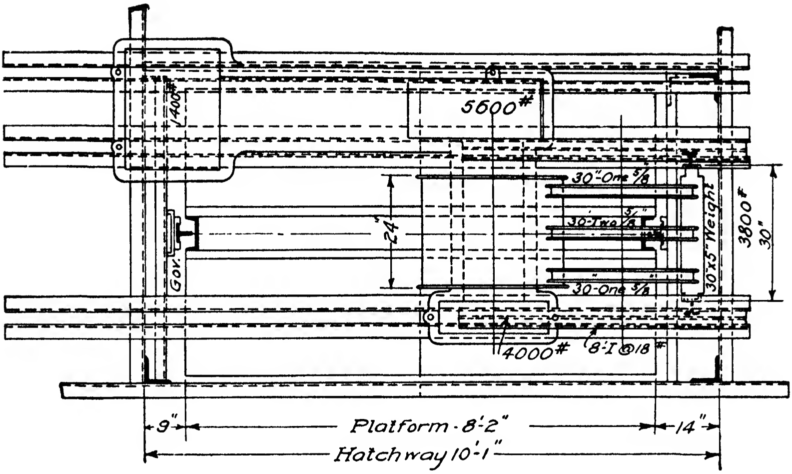



Fig. 208. Overhead Loading for Side-Post Electric Elevator
*Courtesy of Kaestner & Hecht Company, Chicago*

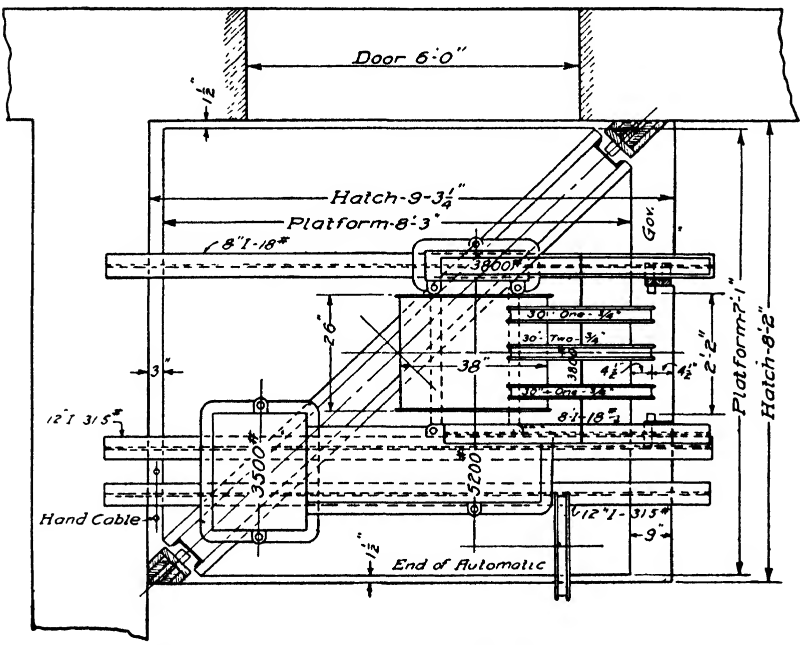
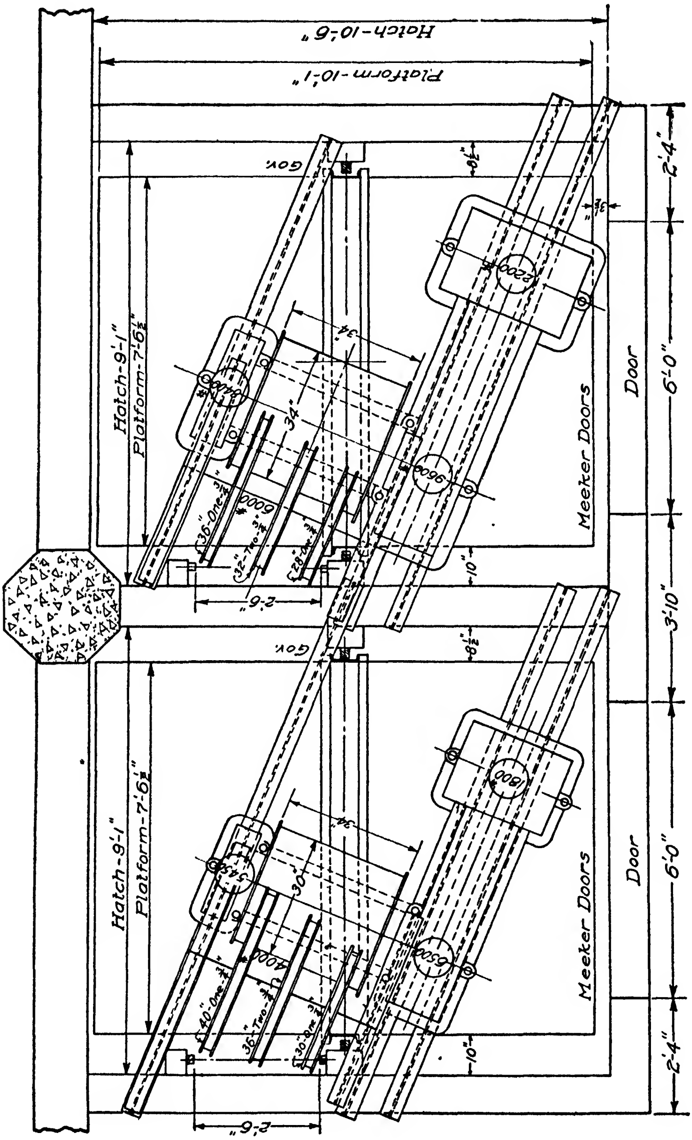



Fig. 209. Overhead Loading for Corner-Post Electric Elevator
*Courtesy of Kaestner & Hecht Company, Chicago*

in case the roof of the penthouse is of glass, but the most popular method is to have a tight floor either of wood or of concrete, as this




HATCH PLAN

Fig. 210 Overhead Loading for Twin Side-Post Elevators

prevents any noise made by the engine from being heard by those riding on the car and also precludes the dropping of oil or of tools

down the hatchway. When a concrete floor is used it is sometimes made strong enough to carry the engine and its load without the use of beams. When a wood floor is used it is made either 3 or 4 inches thick, and the edges of the planks which compose it are tongued and grooved so as to fit together tightly.

*Precautions.* In any case the following precautions must be taken: (1) in estimating the total load on the beams or walls the weight of the floor must be included; and (2) in arranging the engine and its accessories space must be left for a trap door in the floor

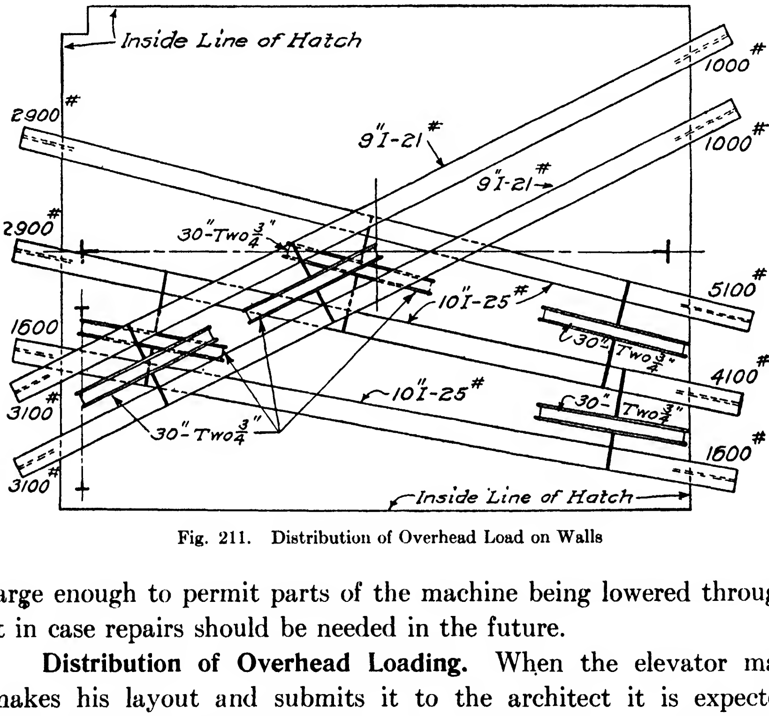



Fig. 211. Distribution of Overhead Load on Walls

large enough to permit parts of the machine being lowered through it in case repairs should be needed in the future.

**Distribution of Overhead Loading.** When the elevator man makes his layout and submits it to the architect it is expected that he will show thereon the dead load and its distribution. For beams carrying only sheaves overhead this is easily done by finding the load each sheave has to carry — as discussed in connection with Fig. 184 — and dividing it between the two boxes or bearings in which the sheave shaft is journaled. But, where the engine is overhead, there must be known in addition to the loads on the sheave bearings the weights of the various parts of the engine and of the magnetic-control switchboard, and, after

finding the loads due to the weight of car, load, and counterpoise, and the points at which they will react, the proportionate weight of that part of the engine which will be carried by the beams at these points must be added, and the distribution of the load shown at these points. This feature is illustrated in Figs. 207 to 210, and 253.

*Calculation by Method of Moments.* Sometimes the architect or engineer who has the erection of the building in hand requires

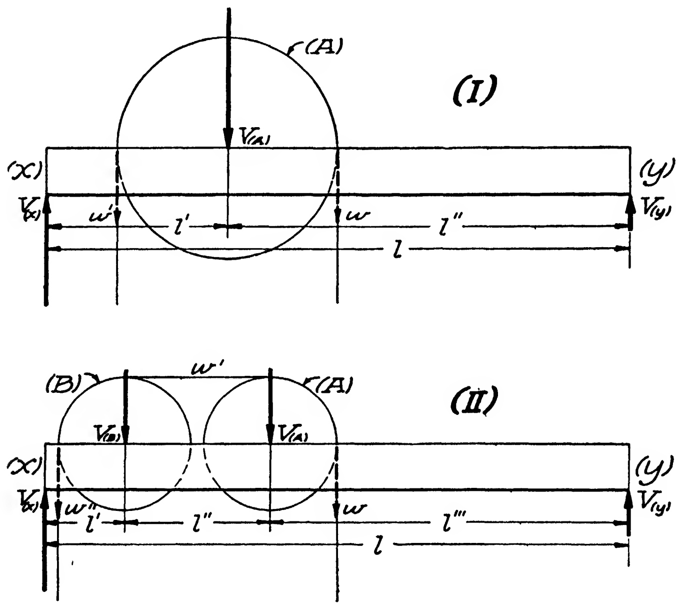



Fig. 212. Diagram of Overhead Beam Loading Conditions

the actual dead loads to be shown on the various points in the walls, as in Fig. 211. Using the method of moments as shown diagrammatically in Fig. 212 — the *moment* about a point being the product of a force and its distance or arm from the point — the vertical reactions $V_{(x)}$ and $V_{(y)}$ at the ends $(x)$ and $(y)$ of a beam may be found as follows: $(I)$ by measuring the distances $l'$ and $l''$ from the wall on either side to the point at which the load or stress $V_{(A)}$ occurs on the beam, and dividing the load proportionally in the inverse ratio of these distances; and $(II)$ where there are two or more loads or stresses $V_{(A)}$ and $V_{(B)}$ at different distances apart $l'$, $l''$ and $l'''$ on

the same beam, the reactions for these at the beam ends are figured separately and the results combined.

Where the load is applied to the beam at one point, as in case (*I*), Fig. 212, the vertical reaction at end ($x$) of the beam may be found by taking the product $V_{(x)}l$ — its moment about the other end ($y$) — equal to the moment $V_{(A)}l''$ of the opposing force or sheave load about the same point ($y$); thus $V_{(x)}=\frac{V_{(A)}l''}{l}$. The vertical reaction at end ($y$) is found similarly and is $V_{(y)}=\frac{V_{(A)}l'}{l}$. The sum of the reactions $V_{(x)}$ and $V_{(y)}$ of course equals their opposing force or load $V_{(A)}$, so in any loading problem of this nature the relation $V_{(A)}=V_{(x)}+V_{(y)}$ may be used in checking the computed reactions.

Where the load is applied to the beam at more than one point as in case (*II*), Fig. 212, the vertical reactions at the beam ends ($x$) and ($y$) may be figured separately for each load and added or may be figured in one operation by the method of equilibrium of moments about each end of the beam. The product, then, of the vertical reaction at end ($x$) and its distance from the other end ($y$) is taken equal to the sum of the opposing moments of the loads at ($A$) and at ($B$) about the same point ($y$); thus $V_{(x)}=\frac{V_{(B)}(l''+l''')+V_{(A)}l'''}{l}$. Similarly, the vertical reaction at end ($y$) is found by equating the moments about ($x$), and is $V_{(y)}=\frac{V_{(A)}(l'+l'')+V_{(B)}l'}{l}$. As before, the sum of these beam reactions must equal that of the applied loads on ($A$) and ($B$), or $V_{(A)}+V_{(B)}=V_{(x)}+V_{(y)}$.

*Illustrative Examples. 1.* Suppose, as in a previous example that a 1200-pound car of 3000-pound capacity — a total static load of 8400 pounds — is supported on one overhead sheave ($A$), as in case (*I*), Fig. 212. Then, if the wheel is 3 feet in diameter, and the supporting beams are 8 feet long, the 8400-pound load is applied $2\frac{1}{2}$ feet from one end and $5\frac{1}{2}$ feet from the other, the reactions being — by the method of moments about opposite ends — 8400 times $5\frac{1}{2}$ divided by 8, or 5725 pounds at the nearer end, and 8400 times $2\frac{1}{2}$ divided by 8, or 2625 pounds at the farther end of the beam. Dividing these by 2 gives the reactions at the ends of each of the twin beams which would be used to support the sheave.

2. If two tandem sheaves are used for the same load, the weight is distributed a little differently on the beams, as in case (*II*), and this is seen to affect the reactions at the beam ends. If two sheaves (*A*) and (*B*), each carrying 4200 pounds vertical static load, have shafts 2 feet from each other and respectively 1 foot and 5 feet from opposite ends of an 8-foot overhead beam, the reactions are found to be, by taking moments about opposite ends, 4200 times 5 plus 4200 times 7 all divided by 8, or 6300 pounds at the nearer end of the beam, and, at the farther end, 4200 times 3 plus 4200 times 1 all divided by 8, or 2100 pounds. These are again divided by 2 for the reactions at the ends of twin supporting beams.

**Selection of Beam Sizes** *Estimate of Wood Beams.* Wood beams for the support of either sheaves or engines are never used in any of the large cities, and they are not allowed there because of the danger of burning in case of fire. There are many places, however, where they are still permitted, and in case it is desired to estimate the required size it can be done by using Table II given in the next section on Elevator Cars.

*Use of Steel-Beam Tables.* The selection of steel beams of proper size to carry the loads imposed on them is usually done by the use of tables, such as Table III in the following section on Car Construction, furnished by the makers of the beams. Carnegie, Jones, and Laughlin, the Cambria, and the Bethlehem steel companies all issue such tables giving minute directions for the proper selection of these beams.

*Allowance for Rigidity.* In all cases the elevator maker should select beams of such size that the impact will not cause them to vibrate in the slightest degree but will remain perfectly rigid under the stress of stopping and starting. It is a good rule therefore to use beams any one of which is amply strong to carry the entire load.

## ELEVATOR CARS

**Essential Elements.** The elemental parts in the construction of elevator cars, of which Fig. 213 is a simple example, are usually considered by elevator makers under two heads: (1) the *platform*, which is the floor and the framework supporting it, and which is the subject we are about to discuss; and (2) the *cage*, which comprises the stiles or upright standards at each side of the car, the crossbeam

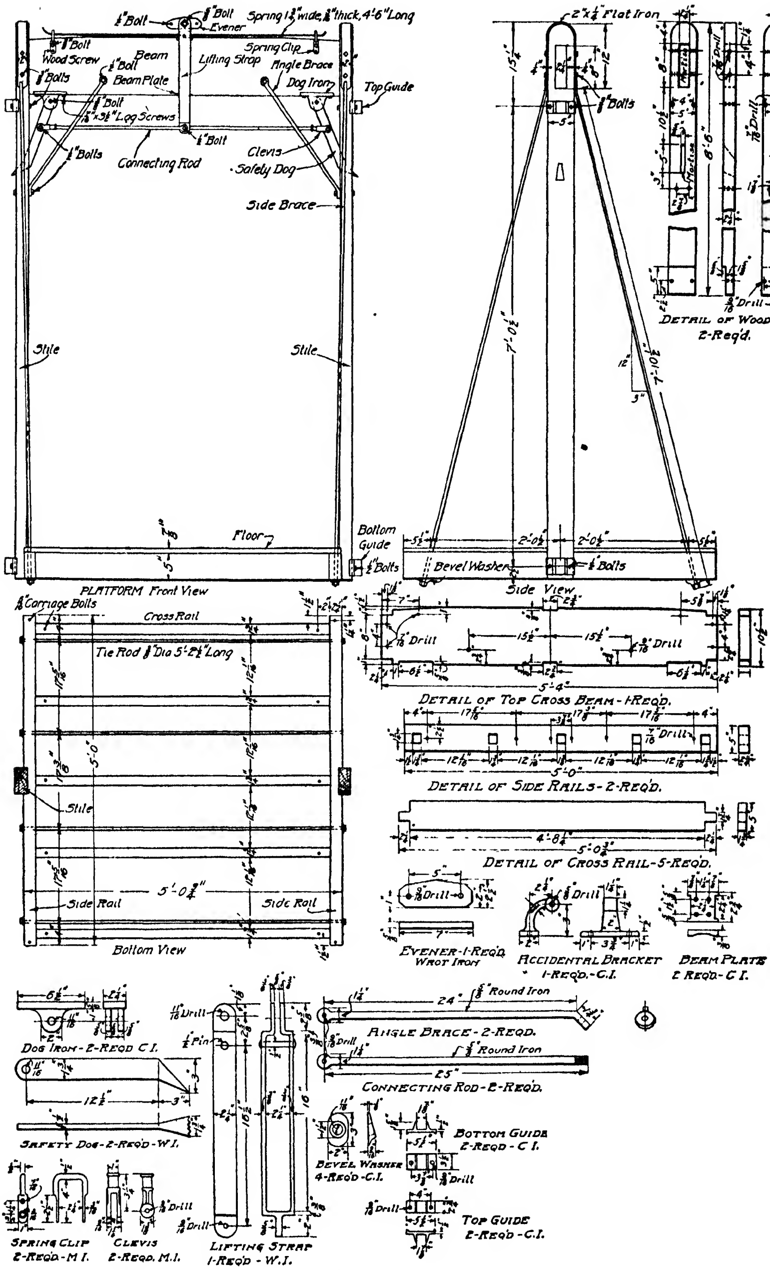



Fig. 213. Elements of Wooden Car Construction Hand-Power Elevator

at the top of the car by which it is lifted, and the bottom beam, or, in lieu of that, the safety plank. Guide shoes are bolted to the top and these shoes travel on the guides which keep the car in its place in the hatchway, as is described later.

## PLATFORMS

**Loading Conditions Affecting Construction.** Lightness and strength combined with safety and durability are the chief qualities to strive for in designing or in building an elevator car. With a

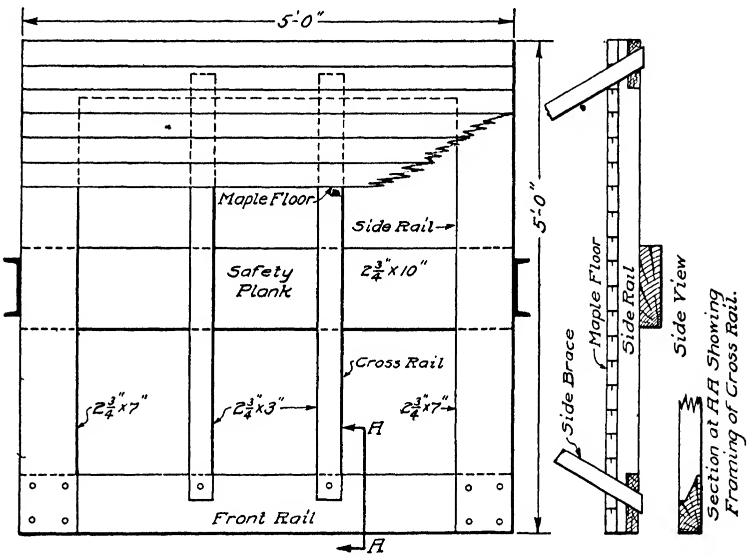



Fig. 214. Flat-Framed Car Platform

small hatchway it is a comparatively easy matter to fill these requirements, especially where the guides are at opposite sides; but when the guides are at diagonally opposite corners, or where the span between the guideposts is great, or the distance from front to rear of hatchway is of considerable length, difficulties multiply, and to a much greater extent than when the span is great both ways. Cars built to run in an abnormally large hatchway usually are intended to carry heavy loads, but where these loads are not contained in one package — that is, where they are placed on the car separately — there is always the danger of piling the goods all on one side, and

hence the car must be built strong enough to carry its full rated load whether it is distributed or is concentrated at one place.

The same difficulties exist where a loaded truck is rolled on the platform, for if the car is stopped a little above or below the landing, this method of loading produces a severe shock either by the wheels striking the front rail, or by dropping from the level of the landing to the floor of the car. Moreover, in such a case the entire load on the truck when rolled onto the platform floor is resting on the four points where the rims of the wheels touch the floor. Unless the platform is built with a view to resisting these peculiar and severe strains—as in the heavy automobile elevator supported at three points, Fig. 223 — it will not last very long. The conditions cited are extreme, but it is by taking note of them that difficulties are foreseen and proper precautions taken to avoid them.

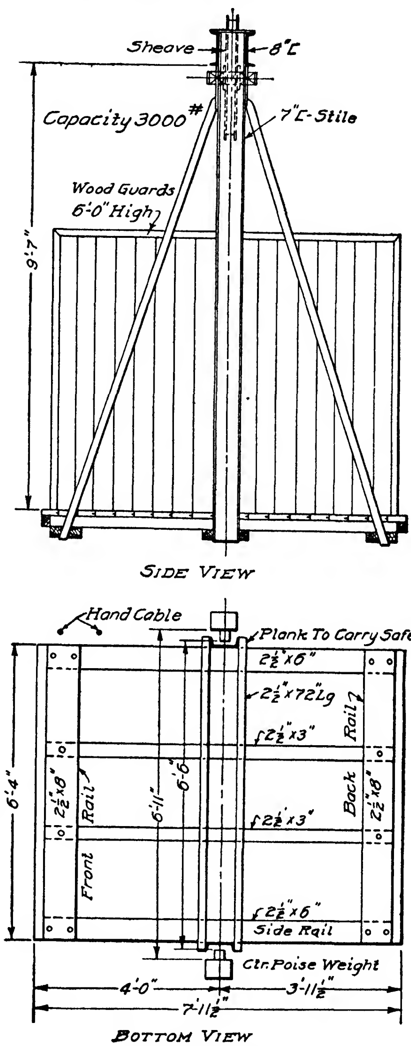



Fig. 215. Economical Type of Flat-Framed Construction

**Hardwood Construction.** Where the platform is made of wood, there are two principal ways of framing the bottom. One is what is called the flat framing, the other is where the members are set on edge in order to get the greatest carrying capacity with the smallest amount of material. By referring to the platforms and cages, Figs. 213 to 227, examples of these two methods of construction may be seen.

In those parts of the country where good hardwood lumber is plentiful very good platforms or cages can be made entirely of hardwood. Of course they have to be ironed, that is, many portions of them have to be made of iron, such as the side braces, lifting strap, angle braces, guide shoes, and the various bolts used for holding the members together and strengthening them. Of course where the standards or stiles are of wood, Figs. 213, 228, and 229, they have to be tenoned into the side rails of the bottom and held there firmly either by iron rods or cheek plates. The crossbeam of the car is

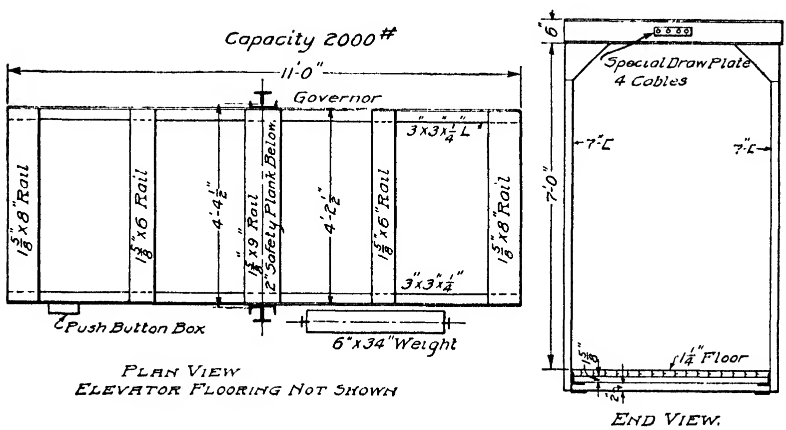



Fig. 216 Combination Wood and Angle-Iron Platform

likewise mortised into the upper ends of the stiles and held there in a similar manner.

*Flat Framing.* Where flat framing is employed for platforms, Figs. 214 to 216, pieces of 3-inch plank dressed to about $2\frac{3}{4}$-inch thickness are used throughout for the rails. Those forming the outer edge of the platform are usually wider than the rails in the center, and the sizes for these vary both with the load to be lifted and the size of the hatchway. Therefore, no rule can be laid down for their dimensions, but some general idea can be given. In the case of a platform 5 feet square which is to carry a load not exceeding 4000 pounds, the outer rails should be 8 or 9 inches wide by $2\frac{3}{4}$ inches thick, and the joints at the corners halved together. The inner rails can be about $2\frac{3}{4}$ by 5 inches. The manner of making the joints with the side rails is shown in the detail of section *AA*, Fig. 214.

In the economical type of construction shown in Fig. 215 no joint fitting is done in building the flat-framed platform, the side and cross rails being simply laid on top of the front and back rails and bolted in place. The stiles and crossbeams are of channel steel.

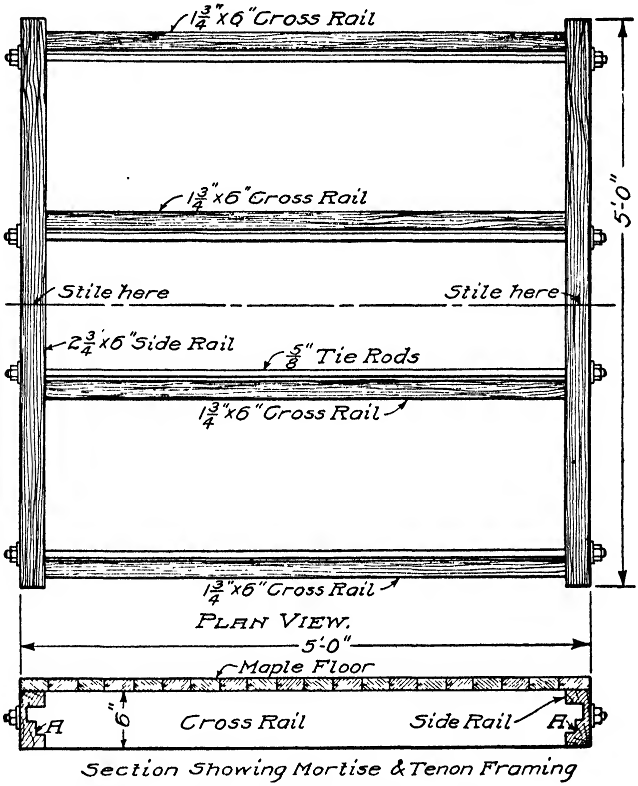



Fig. 217. Edge-Framed Car Platform

The sheave in the crossbeam is used to give the machine a double purchase in lifting.

Another interesting example of economical construction is shown in the 2000-pound capacity car in Fig. 216. The platform is very simply constructed of 3-inch angles for side rails and of $1\frac{5}{8}$-inch plank laid flat for cross rails, while the cage or sling is of 7-inch channels for the stiles which, at the lower ends, are turned under the safety plank.

*Principles of Loading.* As a general principle the platform bottom should be so formed that the side rails carry the entire load,

whatever it may be, the other rails being used simply to carry that portion of it which is allotted to them. The reason why the side rails are the principal factors of strength in the platforms is that they are supported at three points each, namely, at the front and back ends and in the center at the stile. No other portion of the car is so well supported and at so many different points. Hence, these rails are eminently adapted to carry the principal portion of the load. The cross rails, then, carry only a share of it; but, no matter what the load is on the cross rails, the stress is conveyed by them to the side rails, where it is concentrated. The only time when an excessive

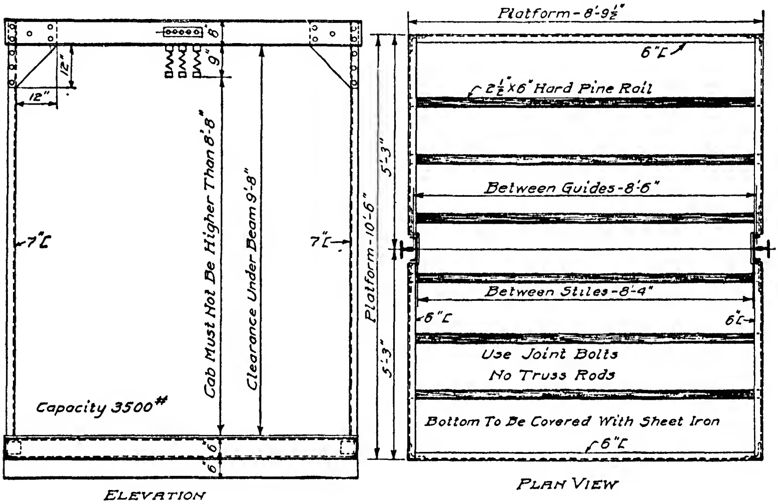



Fig. 218. Steel-Framed Side-Post Car

load is placed on any cross rail is where a loaded truck is wheeled onto the car. Then fully one-half of the load comes on the front rail, but as the truck is rolled farther onto the platform the floor distributes the strain.

Some makers frame the platform bottom in a different way from this, in that all the rails run fore and aft and are framed into the front and back rails. The only support the front and back rails have is what is derived from the side braces, and where the platform is somewhat large the strain is excessive. Evidently this is an objectionable method and one to be avoided.

*Edge Framing.* Where a platform bottom is framed with the rails on edge, Fig. 217, much greater strength may be obtained with

the same amount of material, and no more labor is required in its production. It will be noted in all these cases that the side rails are heaviest, the front and back cross rails being next in strength, and the intermediate cross rails lightest. The reason for this has already been explained.

The method of connecting the side and cross rails is by means of tenons and mortises. The particular form of tenon shown is strongly recommended on account of its giving greater strength with less cutting away of material than where the ordinary tenon and mortise is used. It will be seen by reference to the detail drawing that only a portion of the tenon passes into the side rail, and that the part marked $A$ is much shorter. This gives greater strength to the tenon at the point where the heavy strain comes upon it and at the same time leaves most of the stock in the side rail intact.

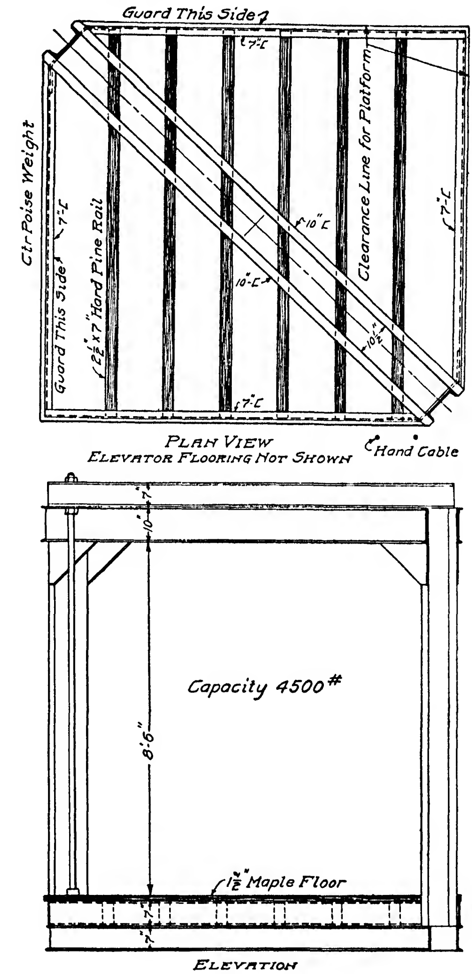



Fig. 219. Steel-Framed Corner-Post Car

**Steel Frames.** Since the introduction of structural steel the making of strong frames has become very much simplified. By using steel channels for both stiles and top crossbeam, and by using two bottom beams made of channel steel also, an exceedingly strong frame can be constructed, as shown in Figs. 218 and 219. This method of construction has many advantages, the most noticeable among them being that

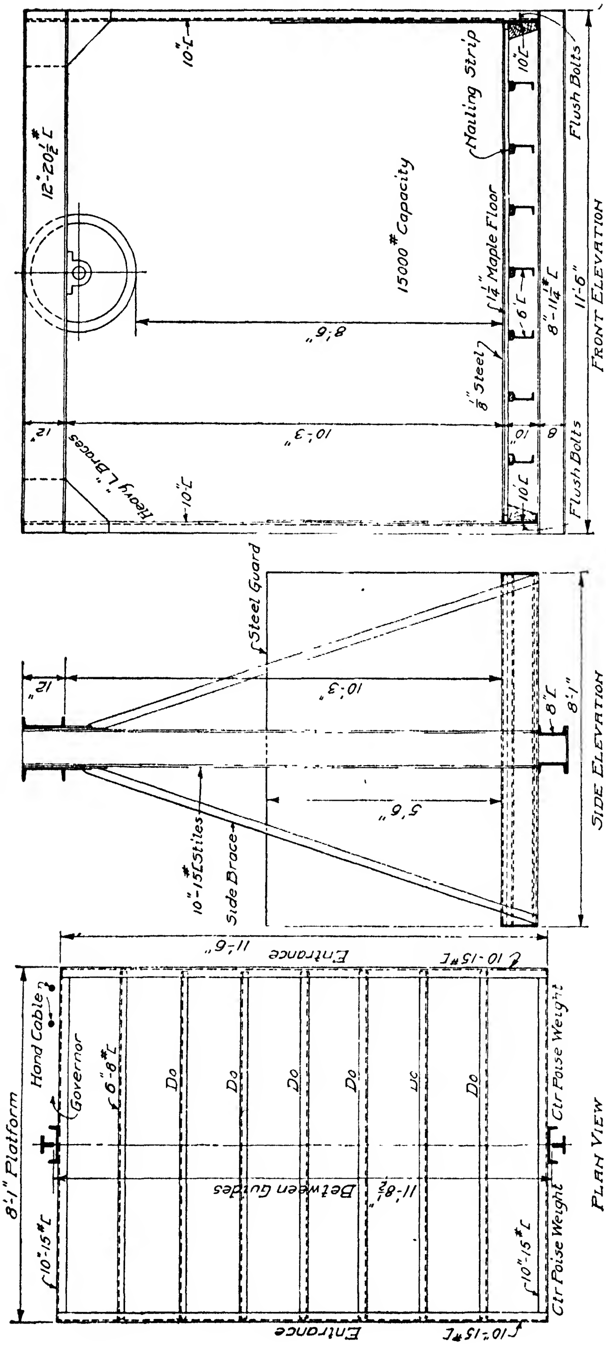



Fig 220 Example of Steel Framing for 15,000-Pound Capacity Car
*Courtesy of Kaestner & Hecht Company, Chicago*

of the simplicity with which a cage which is comparatively long between posts and narrow from back to front may be constructed, as in Fig. 220.

*Platform Construction.* Formerly, when cars were made largely of wood, a platform of the form just referred to was difficult to construct without getting it too heavy. It will be seen by reference to Figs. 218 and 219 that the standards or stiles and the steel frame

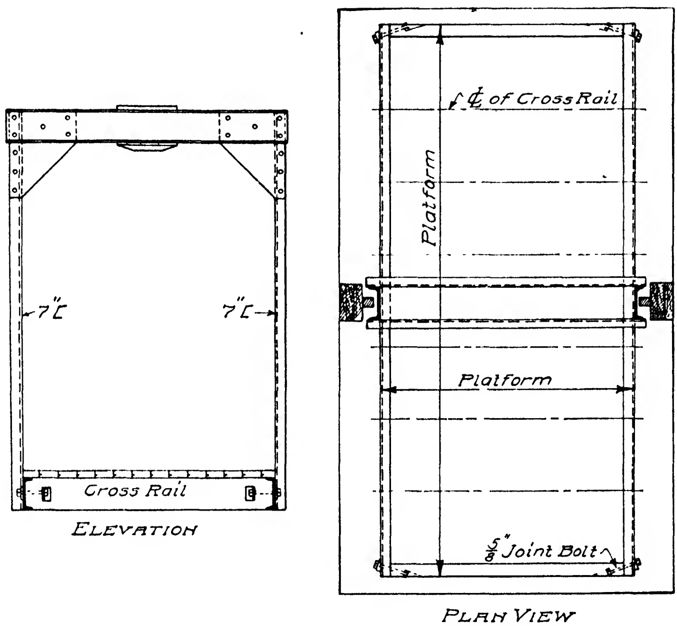



Fig. 221. Use of Joint Bolts in Combination Steel and Wood Platform

composed of the channels forming the upper and lower beams are self-contained so far as strength is concerned, it being only necessary to lay the wood platform upon the lower beams and to support the corners by means of diagonal braces from the stiles. Nothing could be more simple and at the same time more durable. The guide shoes go above and below the upper and lower beams, being bolted to their flanges.

It is only in cases where the platform must be rather long from front to back and comparatively narrow in width that this method of construction seems to lose its advantages. But even here it is more apparent than real, for the only disadvantage is that the side rail is

necessarily made deeper for strength, and that this adds to the depth of car bottom from the floor to the lower point of the guide shoe, which is always an objection. Usually a pit 3 feet deep below the lower landing is the allowance made for elevator cars, but this method when applied to platforms of great depth from front to back necessitates a deeper pit, which cannot be always obtained. In such a case the bottom beams may be dispensed with by bolting the stiles to the side rails, Fig. 221, and by using channels to form the latter.

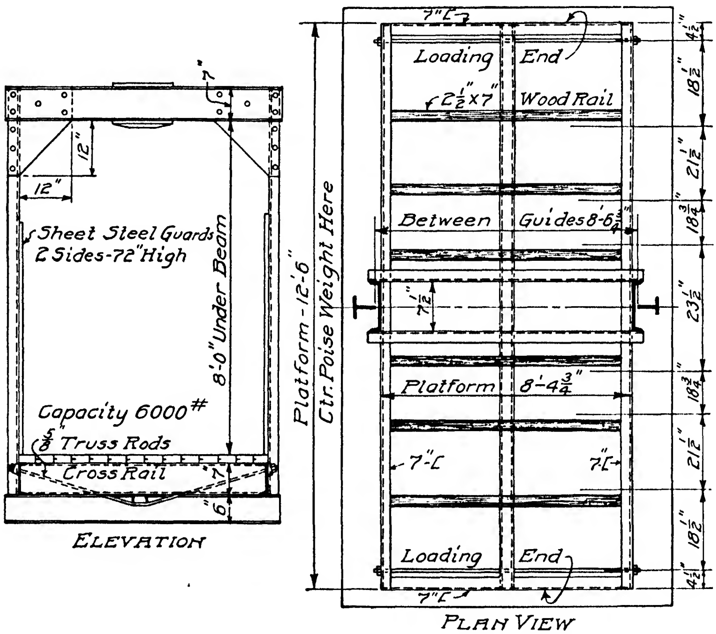



Fig. 222. Use of Truss Rods in Combination Steel and Wood Platform

With this form of construction either wood cross rails or those made of channels may be used, the determination of this detail depending somewhat on the load to be carried. If wood cross rails are used, their ends have to be fitted to the inside of the channel, the lower flange of which carries the cross rail. Either joint bolts or tie rods may be used to hold the frame together. If joint bolts are used, Figs. 218 and 221, not less than one in each end of each cross rail should be used. If tie rods are employed, as in Fig. 217, there should be one across the side of every cross rail. The floor of the platform,

which is made of 1¼- or 1½-inch maple, is nailed to these cross rails and holds them in position. Should it be found that the rails require reinforcing, it can be done by making the tie rods act as truss rods, as shown by Figs. 222 and 223.

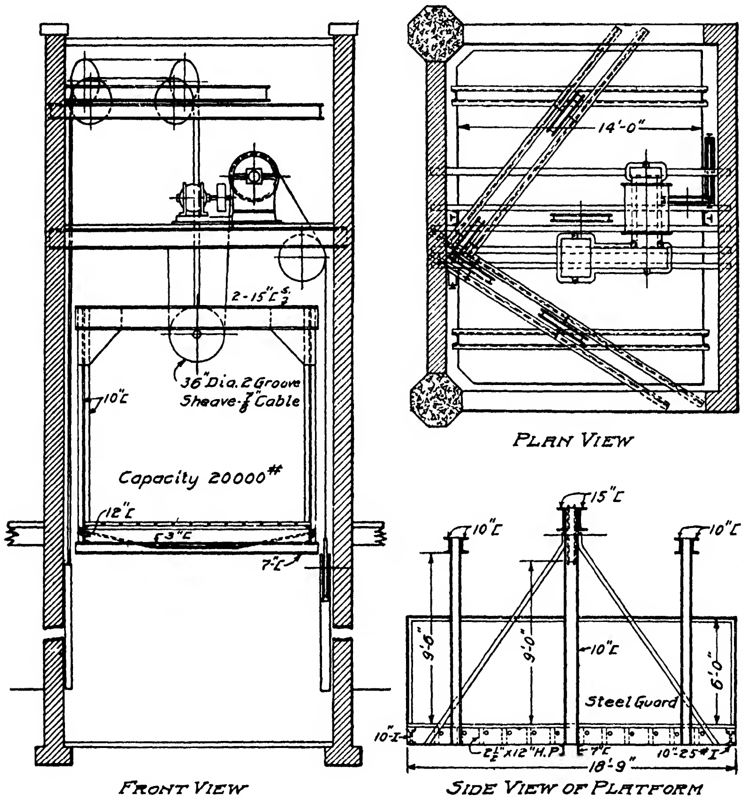



Fig. 223. 20,000-Pound Capacity Double-Counterpoised Automobile Elevator
*Courtesy of Kaestner & Hecht Company, Chicago*

### Proportioning Platform Members

**Hardwood Platform Rails.** When of wood and for loads ranging from 3000 to 6000 pounds, the side rails are usually of 3-inch stock dressed to 2¾ inches, as already referred to. The front and back cross rails are of the same thickness. The intermediate cross rails are made from either 2- or 2½-inch stock dressed to 1¾ or 2¼ inches, respectively,

## TABLE II

### Safe Loads for Wood Beams

CALCULATED ON THE BASIS OF 750 POUNDS PER SQUARE INCH EXTREME FIBER STRESS, CORRESPONDING TO THE FOLLOWING VALUES FOR MODULI OF RUPTURE

Spruce and White Pine, 3000 Pounds; Oak, 4000 Pounds; Yellow Pine, 5000 Pounds

| BEAM DEPTH (in.) | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| SPAN (ft.) | SAFE LOAD (lbs.) | | | | | | | | | | |
| 5 | 600 | 820 | 1070 | 1350 | 1670 | 2020 | 2400 | 2820 | 3270 | 3750 | 4270 |
| 6 | 500 | 680 | 890 | 1120 | 1390 | 1680 | 2000 | 2350 | 2730 | 3120 | 3650 |
| 7 | 430 | 580 | 760 | 960 | 1190 | 1440 | 1710 | 2010 | 2330 | 2680 | 3050 |
| 8 | 380 | 510 | 670 | 840 | 1040 | 1260 | 1500 | 1760 | 2040 | 2340 | 2670 |
| 9 | 330 | 460 | 590 | 750 | 930 | 1120 | 1330 | 1560 | 1810 | 2080 | 2370 |
| 10 | 300 | 410 | 530 | 670 | 830 | 1010 | 1200 | 1410 | 1630 | 1880 | 2130 |
| 11 | 270 | 370 | 490 | 610 | 760 | 920 | 1090 | 1280 | 1490 | 1710 | 1940 |
| 12 | 250 | 340 | 440 | 560 | 690 | 840 | 1000 | 1180 | 1360 | 1560 | 1780 |
| 13 | 230 | 310 | 410 | 520 | 640 | 780 | 930 | 1080 | 1260 | 1440 | 1530 |
| 14 | 210 | 290 | 380 | 480 | 590 | 720 | 860 | 1010 | 1170 | 1340 | 1530 |
| 15 | 200 | 270 | 360 | 450 | 560 | 670 | 800 | 940 | 1090 | 1250 | 1420 |
| 16 | 190 | 260 | 330 | 420 | 520 | 630 | 750 | 880 | 1020 | 1180 | 1330 |
| 17 | 180 | 240 | 310 | 400 | 490 | 590 | 710 | 830 | 960 | 1100 | 1260 |
| 18 | 170 | 230 | 290 | 370 | 460 | 560 | 670 | 780 | 910 | 1040 | 1190 |

NOTE.—Table II gives the safe load for a beam 1 inch thick, of the depth and distance between supports given, for white pine or spruce. To find the safe load for any wood beam of given thickness and depth, multiply the safe load given in the table by the thickness of the beam in inches and fractions of an inch. For oak, increase the values in table by $\frac{1}{3}$; for yellow pine, by $\frac{2}{3}$.

the depth being from 5 to 9 inches, according to the length of rail and the load it has to carry.

**Safe Loads for Wood Beams.** The method of arriving at the proper proportions for these rails is by means of Table II, which has been compiled as a result of numerous experiments to determine the strength of wood beams of the woods chiefly used in mechanical and architectural structures. This table is based upon a factor of safety of 5 and gives the safe loads for beams 1 inch thick and of various depths and spans.

*Use of Table.* The safe load given in the table for a beam of known depth and span, when used as a divisor of the load the rail has to carry, gives the proper thickness for the rail decided upon. Or, the thickness of rail may be assumed; in which case the load to be

placed on that rail is divided by the assumed thickness in inches, and the result will be the safe load for the beam of that span, but only 1 inch thick. Hence, by referring to the horizontal column for that span in the table, the depth of the beam can be determined.

The method of determining the amount of load each rail will carry is not very easily defined by rule, and reliance must be placed somewhat on keen foresight or on past experience, care being taken to allow a safe margin.

**Interdependence of Platform Parts.** *Side Rails.* In the case of a platform with the guides at the sides, the side rails will have to carry the entire load between them, including the cross rails and floor. They will also be weakened somewhat by the mortises cut into them for the attachment and support of the cross rails. On the other hand, they will be strengthened by the support of the stiles in the center and the side braces at each end. Hence, if the mortises are made in a proper manner, it will be safe to consider the side rails as being beams of the thickness and depth determined upon for them and supported at their extreme ends.

*Cross Rails.* The front cross rail, which is located at that part of the platform where the loading and unloading is done, and the back cross rail also, if loading is done there, too, should be of the same dimensions as the side rails. The intermediate cross rails may be lighter. Usually these cross rails are placed from 18 to 24 inches apart on centers, according to the way in which the platform can be divided from back to front. Care should be taken to use an even number of rails in order to avoid having a rail running across between the stiles, for it is better to have one rail on each side of the stiles. The guide shoe is then set on a short plank bolted to the two middle cross rails. The floor, being laid on top across the cross rails from back to front, gives its strength to support the load; and all this must be taken into consideration in estimating the load the cross rails have to carry.

**Platform Construction Problem.** *Wooden Side Rail.* Considering the side rails first, let it be assumed, for example, that an 8-foot square platform is to be built to carry 4000 pounds. For experimental calculation the rails will be assumed to be 7 inches deep. By reference to Table II we find that the safe load for a pine rail 7 inches deep,

1 inch thick, and of 8-foot span is 510 pounds. This side rail is to be of 3-inch hardwood, either oak or maple, dressed to $2\frac{3}{4}$ inches. If the rail were of white pine or spruce, it could carry 510 pounds times $2\frac{3}{4}$, or 1402 pounds, but being of hardwood, we may allow fully $\frac{1}{3}$ more load on it — the total allowance being 1402 times $1\frac{1}{3}$, or 1870 pounds. This is the load the rail on edge will carry if supported only at the ends; but it is supported in the center by the stile and at the ends, making three supports less than 4 feet apart, and this doubles its strength, making it 3740 pounds. However, the mortises have a weakening effect, so it is well to assume only $\frac{2}{3}$ of this value, or, say 2500 pounds, which is $\frac{5}{8}$ of the load the platform is to carry.

*Wooden Cross Rails.* The front and back rails will carry safely 1402 pounds each. These rails are framed into the side rails at a distance of $1\frac{1}{4}$ inches from the ends, thus making the distance between the inside surfaces of front and back rails equal to 8 feet minus both twice $1\frac{1}{4}$ inches and twice $2\frac{3}{4}$ inches, or 89 inches. This divided by 5 will allow 4 inside cross rails about $17\frac{1}{2}$ inches between centers. Dividing 4000 pounds by 6, the total number of cross rails, gives less than 700 pounds as the weight each cross rail has to carry. From Table II it is found that an 8-foot rail 7 inches deep by $1\frac{3}{4}$ inches thick will safely carry 510 times $1\frac{3}{4}$ times $1\frac{1}{3}$, or 1190 pounds, if made of hardwood. Therefore, all these rails will be amply strong, except possibly the front and back rails in case the load should be placed on a truck and rolled onto the platform. In this case fully one-half of the load comes on the front rail for a moment, and there is the possible shock from the platform not being stopped even with the floor before loading. It would seem as if the safe thing to do would be to truss this front rail, but as yet the actual conditions under which these cross rails will work have not been considered.

*Car Flooring.* There is the floor as yet unconsidered. This should be made of $1\frac{1}{4}$-inch stock dressed to $1\frac{1}{16}$ inches and used in widths of 5 inches. A floor of this thickness made of maple, when unsupported between the front and back rails, will carry for the same width $\frac{1}{16}$ of what each inside cross rail will carry. Allowing a good margin for safety, it may be safely assumed that the floor when unsupported in the center will safely sustain of itself a load of 20 pounds to the square foot, that is, less than half the amount figured by the table. The area of the floor being 8 times 8 feet, or 64 square

feet, a total load of 64 times 20 pounds, or 1280 pounds, can be sustained by the floor itself. Hence the crossbeams are only called upon to support 4000 minus 1280 pounds, or 2720 pounds.

But these considerations must not enter into the estimate of the required strength of the cross rails in determining their dimensions, but should be taken into account simply for the purpose of becoming fully acquainted with the actual physical conditions of the problem. This additional strength contributed by the floor may be considered a reserve against the piling of heavy goods in one place on the platform, or the rolling of a heavy safe thereon, or any other severe usage.

*Truss Rod for End Rail.* In order to use as light a truss rod as possible, it will be found best to use a double strut truss under this rail. This will divide the rail into 3 parts, the load on each division to be carried by the truss being $\frac{1}{3}$ the surplus load on the rail, which

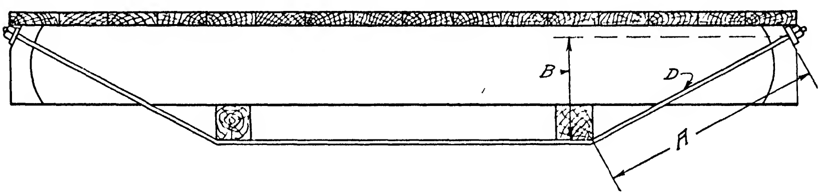



Fig. 224. Diagram of Trussed Rail

was figured to carry safely alone 1402 pounds. Hence, if it is desired to so truss the rail that it will safely carry $\frac{1}{2}$ of the total load the platform is to carry, the surplus will be 2000 minus 1402, or 598 pounds, $\frac{1}{3}$ of which is, say, 200 pounds.

Referring to Fig. 224, the tension on *D* is equal, practically, to 200 pounds times *A* divided by *B*. The distance *B* is 7 inches and the distance *A* is 33 inches. Hence the tension on rod *D* is 200 pounds times 33 divided by 7, or 943 pounds. The tensile strength of wrought iron is 50,000 pounds per square inch. Hence, with a factor of safety of 5, it may be subjected to a tensile stress of 10,000 pounds per square inch. However, this rod is only called upon to carry a stress of 943, or about $\frac{1}{10}$ of 10,000 pounds. Hence its area could be about $\frac{1}{10}$ of a square inch. A rod $\frac{3}{8}$ inch in diameter has an area a little greater than $\frac{1}{10}$ of a square inch, but, to be perfectly safe, a $\frac{5}{8}$-inch rod should be used.

# TABLE III

## Safe Loads Uniformly Distributed for Standard and Special Channels

(Carnegie Shapes)

| Supports Spaced (ft.) | Capacity (Net tons, 2000 lbs.) (*Lbs. additional for each pound increase in weight of section) | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | * | | * | | * | | * | | * | | * | | * | | * | | * |
| 5 | 2.02 | 21 | 3.16 | 26 | 4.62 | 31 | 6.68 | 36 | 8.61 | 42 | | | | | | | | |
| 6 | 1.68 | 18 | 2.63 | 22 | 3.85 | 26 | 5.57 | 30 | 7.18 | 35 | | | | | | | | |
| 7 | 1.44 | 15 | 2.26 | 19 | 3.30 | 22 | 4.77 | 26 | 6.15 | 30 | | | | | | | | |
| 8 | 1.26 | 13 | 1.98 | 16 | 2.89 | 19 | 4.18 | 23 | 5.38 | 26 | | | | | | | | |
| 9 | 1.12 | 12 | 1.76 | 14 | 2.57 | 17 | 3.71 | 20 | 4.78 | 23 | | | | | | | | |
| 10 | 1.01 | 11 | 1.58 | 13 | 2.31 | 16 | 3.34 | 18 | 4.31 | 21 | 5.61 | 24 | 7.14 | 26 | 11.39 | 32 | 22.23 | 39 |
| 11 | .92 | 10 | 1.44 | 12 | 2.10 | 14 | 3.04 | 16 | 3.91 | 19 | 5.10 | 21 | 6.49 | 24 | 10.35 | 29 | 20.20 | 35 |
| 12 | .84 | 9 | 1.32 | 11 | 1.93 | 13 | 2.78 | 15 | 3.59 | 18 | 4.68 | 20 | 5.95 | 22 | 9.49 | 26 | 18.52 | 33 |
| 13 | .78 | 8 | 1.22 | 10 | 1.78 | 12 | 2.57 | 14 | 3.31 | 16 | 4.32 | 18 | 5.49 | 20 | 8.76 | 24 | 17.10 | 30 |
| 14 | .72 | 8 | 1.13 | 9 | 1.65 | 11 | 2.39 | 13 | 3.08 | 15 | 4.01 | 17 | 5.10 | 19 | 8.14 | 23 | 15.87 | 28 |
| 15 | | | | | | | | | | | 3.74 | 16 | 4.76 | 17 | 7.59 | 21 | 14.82 | 26 |
| 16 | | | | | | | | | | | 3.51 | 15 | 4.46 | 16 | 7.12 | 20 | 13.89 | 24 |
| 17 | | | | | | | | | | | 3.30 | 14 | 4.20 | 15 | 6.70 | 18 | 13.07 | 23 |
| 18 | | | | | | | | | | | 3.12 | 13 | 3.96 | 14 | 6.33 | 18 | 12.35 | 22 |
| 19 | | | | | | | | | | | 2.95 | 12 | 3.76 | 14 | 5.99 | 17 | 11.70 | 21 |
| 20 | | | | | | | | | | | 2.81 | 12 | 3.57 | 13 | 5.70 | 16 | 11.11 | 20 |
| Channel Depth (in.) | 4 | | 5 | | 6 | | 7 | | 8 | | 9 | | 10 | | 12 | | 15 | |
| Weight (lb. per lin. ft.) | 5.25 | | 6.5 | | 8 | | 9.75 | | 11.25 | | 13.25 | | 15 | | 20.5 | | 33 | |

*Careful Framing Required.* Care should be had in framing and putting the platform together to make the tenons and mortises as shown in Fig. 217, and to bind the whole together with either joint bolts or truss rods.

**Steel and Combination Platforms.** *Use of Channels.* The rules and the example given here may be applied to the framing of any platform constructed in this manner, whether of wood or steel, except that in the use of steel channels for any or all the rails reference must be made to some standard table of safe loads for channels. Table III gives the Carnegie Steel Company's tabulation.

An exceedingly strong and durable platform is frequently made by using steel channels for side rails and wood for the cross rails. The cross rails may or may not be trussed, as found to be necessary by an analytical computation such as outlined in the foregoing example. It must be borne in mind, however, that where wood cross rails and steel side rails are used it frequently becomes necessary, in order to be able to use cross rails of a proper depth, to use side rails of a greater depth than would be really necessary for strength. Proportionate strength in cross rails is not obtainable by making them of greater thickness without waste of material and additional and unnecessary weight of platform. This latter feature is one to be avoided always, because, besides the waste of material directly involved, an abnormally heavy platform causes further waste by making the use of additional counterpoise weights imperative in order to offset the excessive weight of the platform. There is still another undesirable feature introduced by excess of material in increasing the inertia to be overcome in starting and stopping the elevator.

*Attachment of Floor.* When the platform, except the floor, is made entirely of steel, the cross rails must be coped to fit the side rails, and the attachment of the floor may be accomplished by drilling the upper flanges of both side and cross rails at intervals for the insertion of bolts which pass through both floor and flanges, either small carriage or stove bolts being used, with nuts below.

Another way is to use side rails of somewhat greater depth and to bolt pieces of hardwood along the top flanges of the cross rails to nail the floor to, Figs. 220 and 223. In this method of construction

small I-beams may be used instead of channels for cross rails if greater strength is desired, or they may be trussed if it is found best to do so.

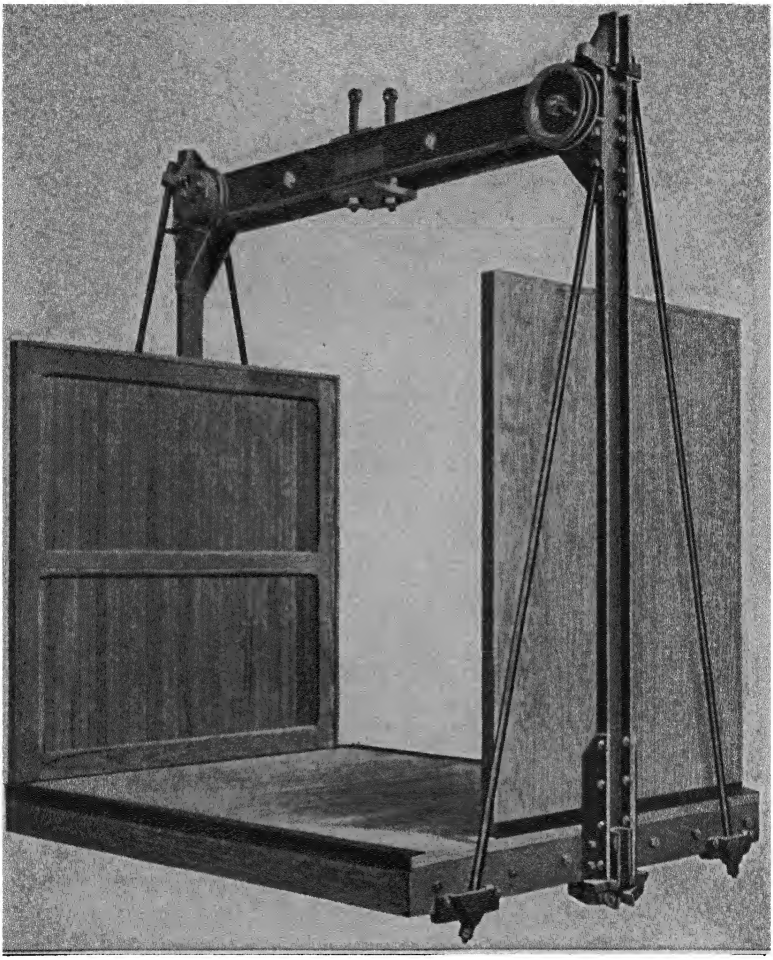


**Fig. 225. Common Type of Freight Car with Wood Guards**
***Courtesy of Otis Elevator Company, New York City***

### Guards

**Typical Features.** Guards are generally placed on all sides of the car not used for loading and unloading. They are sometimes of wood, Fig. 225, but more frequently of sheet metal, Figs. 220 and

223. Sometimes they are made of flat bar iron, $\frac{1}{4}$ by $1\frac{1}{4}$ inches, in the form of latticework as shown in Fig. 226, but this type is not a favorite because of the possibility of small packages slipping through the openings. They are also made of strong wire mesh

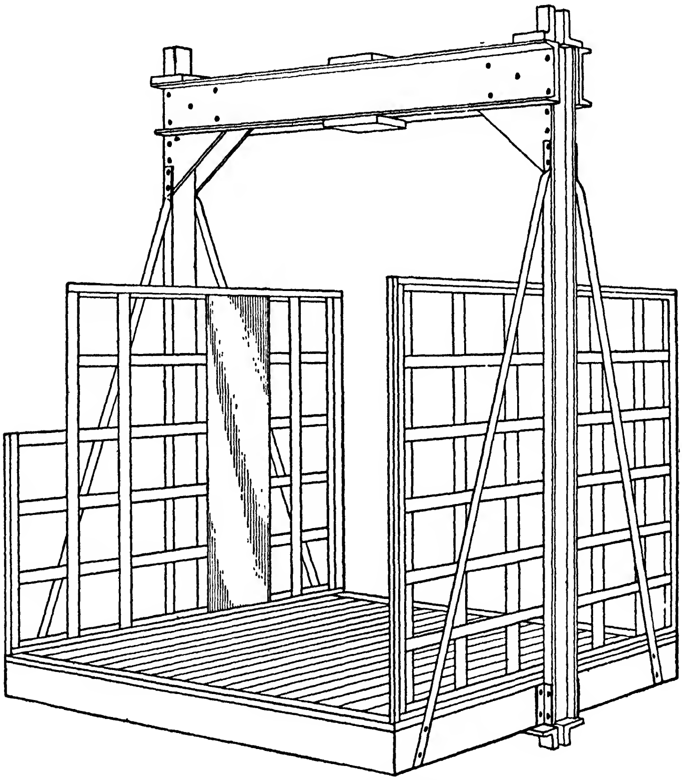


Fig. 226. Freight Car with Lattice Guards
*Courtesy of Kaestner & Hecht Company, Chicago*

fastened to channel frames, as in Fig. 227. Such guards are usually made of No. 8 or 10 wire, being $1\frac{1}{4}$ to $1\frac{1}{2}$-inch diamond-shaped mesh.

The last two arrangements are open to the objection of not being entirely safe because they do not form a perfect safeguard against the passing counterpoise weight. The solid wood guard or the sheet steel are the best and safest. No guards should be less than 5 or 6 feet high, so that employes riding on the elevator cannot rest their elbows on the top of the guard and thus endanger their

personal safety by being caught by the passing counterpoise weight or by other projections in the hatchway.

**Wooden Construction.** When made of wood, the guards are usually of $\frac{7}{8}$-inch wainscoting — that is, of 1-inch boards from $2\frac{1}{2}$ to 6 inches wide, dressed on both sides and fitted together by tongues

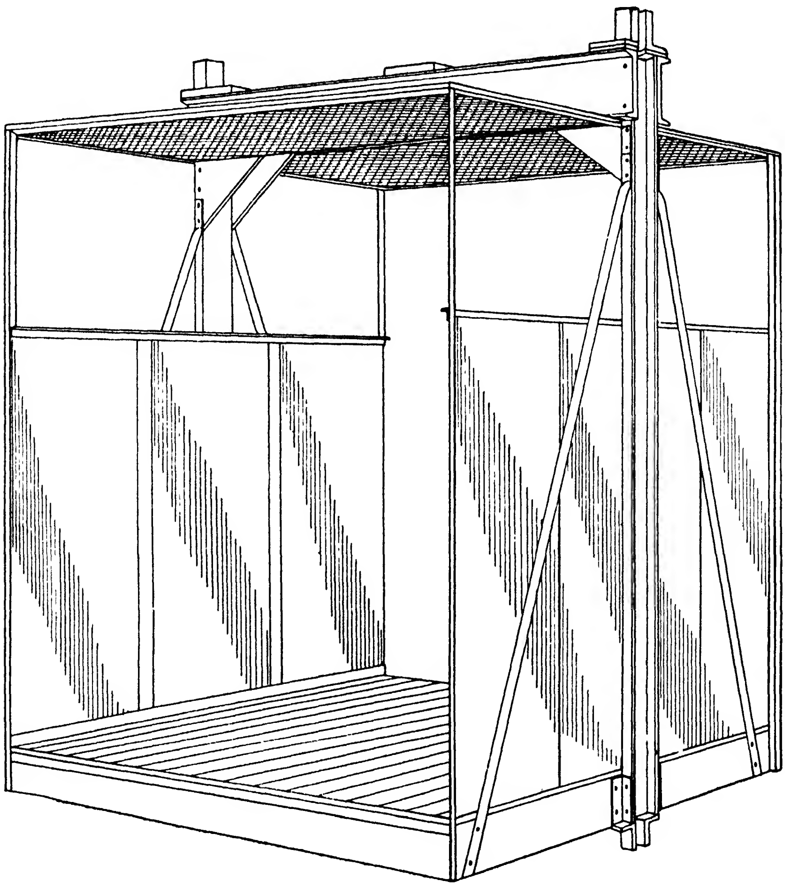


Fig. 227. Freight Car with Sheet-Steel Guards and Wire Screen
*Courtesy of Kaestner & Hecht Company, Chicago*

and grooves. The upper ends are capped, and the lower edge of the wainscoting is attached to a strip screwed to the floor of the platform and to the stiles and side braces. This makes a comparatively cheap and stiff guard, but it will not wear as long as those made of sheet steel.

**Sheet-Steel Guards.** In the sheet-metal construction the guards are usually made of No. 14 and No. 16 steel sheets, riveted and stiffened by angle-iron frames. They are fastened to the floor of the car by screws passing through the angle iron which forms the bottom of the frame, and are also attached to the stiles and side braces by bolts or screws, a heavier angle iron being used on the back and front edges. This angle is made long enough to pass down below the floor, where it is bolted to the side rail. The angle forming the frame proper is about 1 by 1 by $\frac{3}{16}$-inch, while those which form the stiffeners at the outer edges are usually 2 by 2 by $\frac{3}{16}$ inches.

## CAGE CONSTRUCTION

### Stiles

**Construction Conditions.** When the car is made of wood the stiles or standards are stout planks from $2\frac{1}{2}$ to $4\frac{1}{2}$ inches thick according to the service to which the elevator is to be put, Figs. 213, 228, and 229. When of steel, Figs. 215 and 216, 218 to 223, 225 to 227, and 232, they are usually made of 5-, 6-, or 7-inch channels for light or ordinary work, but for heavy duty they are frequently made of 9-, 10-, or even 12-inch channels. Standard-weight sections are sufficient for most purposes, but, when the platform is very large or the length of the stile great, special channels with extra thick backs are used, or I-beams are substituted. Owing to the use of side braces and the fact that the tensile strength of iron and steel is great, no calculations for tensile strength are necessary either in the case of wood or of steel — for in the case of wood stiles the side braces insure ample strength in that direction — but it is the side strains tending to bend the stile which have to be guarded against. The methods of attaching the stiles to the top beam and to the bottom beam, or to the bottom frame or platform where no bottom beam is used, are very important points to consider.

**Angle Braces for Stiles.** In attaching the stiles to the top beam, angle braces usually are used across the corners where the stiles join the beam, to keep the beam and the stiles square with one another. With wood construction these braces are usually of wrought iron, Fig. 228, though some prefer a cast-iron angle piece, Fig. 257, at each corner. With steel construction, Figs. 218 to 227, gusset plates or triangles of flat sheet steel are used.

**Attachment of Stiles to Platform.** The manner of attaching the stiles to the platform differs with the construction of the car. Where a bottom beam is used, the wood platform is simply notched out at the stiles, which are then bolted to the bottom beams. The notching out prevents the possibility of the platform moving horizontally in

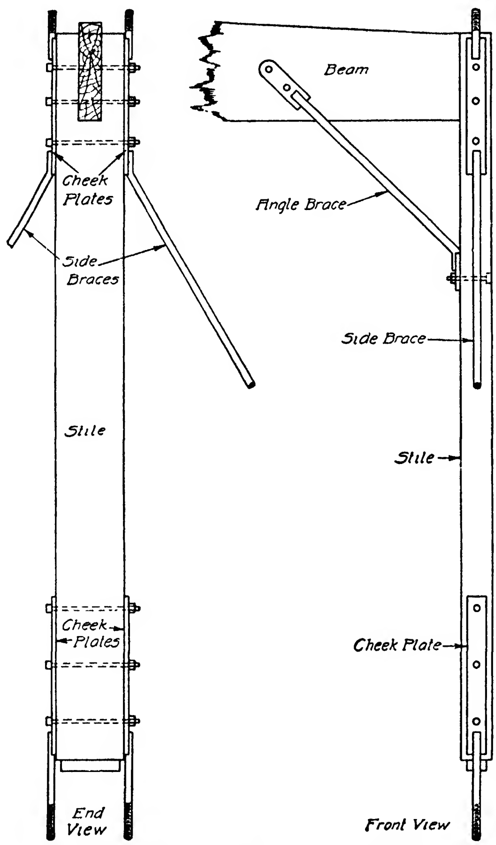



Fig. 228. Details of Wooden Stile for Side-Post Car

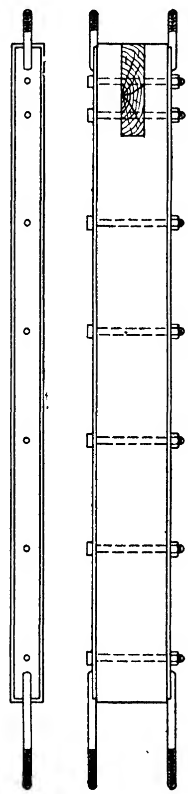


Fig. 229. Wooden Stile for Corner-Post Car

either direction. The side braces extending from up near the top beam diagonally to the front and back corners support the front and back of the platform when loaded. Where cars have a framed bottom, or where only a safety plank is used, the stiles are bolted at their bottom ends to the platform, as shown in Fig. 236, (*b*). This plan can be very substantially carried out when the side rails are moderately deep.

**Cheek Plates for Stiles.** In wooden construction the stiles are reinforced by the use of cheek plates of iron or steel, as shown in Fig. 228. These are flat plates, usually 2 by $\frac{1}{4}$ or $\frac{3}{8}$ inches, formed at one end in the shape of a bolt of $\frac{5}{8}$-, $\frac{3}{4}$-, or $\frac{7}{8}$-inch diameter. These round bolt ends are made long enough to go through the platform, safety plank, and guide shoes and nuts, thus securely fastening the stiles and guide shoes. The cheek plates which are used at the upper ends of the stiles do not require such long studs, as they only pass through the upper guide shoes, which are fastened in place similarly to the lower shoes.

Where the guides are at the corners of the car, the stiles have to do the entire lifting at those points, and accordingly the cheek plates are made to extend the entire length of the stile. In the case of side guides, the chief dependence for lifting is placed in the side braces, so the cheek plates are made to extend only 16 to 20 inches along the stiles.

**Use of Side Braces with Stiles.** The side braces are sometimes made of round iron, in other cases of flat bars, according to the ideas of the maker. When of round iron they are from $\frac{5}{8}$ to 1 inch in diameter. For wood construction they are made a part of the upper cheek plate and their lower ends are threaded and passed through the platform bottom, as shown in Fig. 213. Nuts are used at their ends to carry the load, these nuts being also used, when installing the car, to adjust the bottom to a proper level.

*Attachment of Flat Braces.* When the side braces are of flat iron, which is usually the case in steel construction, the sizes used vary with the size of the car and the load it has to carry. The most common sizes are $1\frac{1}{2}$ by $\frac{3}{8}$ inch, 2 by $\frac{1}{2}$ inch, and $2\frac{1}{2}$ by $\frac{5}{8}$ inch. The upper end of the brace is bolted just below the top beam to the flange of the channel which forms the stile. The bar is then twisted $\frac{1}{4}$ turn, thus bringing the flat side into position to bolt to the side rail of the platform. Two bolts are used at each end, and are usually $\frac{5}{8}$ inch in diameter, but in extra heavy cars bolts of $\frac{3}{4}$-inch diameter may be found essential. With this type of side braces no adjustment is possible, so that they must be accurately fitted in the making of the car.

*Attachment of Round Braces.* In making use of a side brace of round iron, Fig. 225, some makers attach the lower end through a bracket of cast iron bolted to the side rail. The bracket has a shelf

through which the round side brace passes, and against which the nut bears.

*Stiffness Regulates Size.* Side braces made as described are always amply strong so far as tensile stresses are concerned. This can readily be proved by figuring the area in square inches at the smallest part of the brace, and by estimating the resistance to shearing stress offered by the bolts which fasten them to the stile or side rail. The chief consideration in determining the proper size of the braces is their stiffness, a slim spring brace being undesirable even though its section shows it to be ample as regards tensile stress.

## Crossbeams

**Design of Main Beam.** *Essential Conditions.* A very important member of the car is the top crossbeam. In estimating the proportions for this beam, two essential features must be kept in

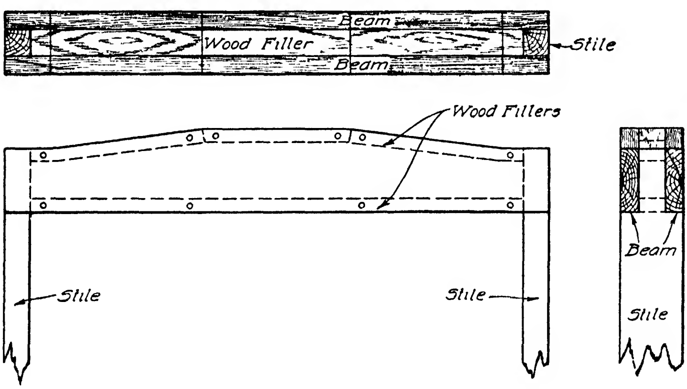



Fig. 230. Construction of Double Wood Beam

mind: (1) that this beam has to carry, not only the load, but also the weight of the car itself; and (2) that this stress is applied at the center of the beam.

*Computation of Size.* Tables II and III are based on loads evenly distributed over the length of the beam between supports. When the load is all at or near the center, it is customary to estimate on a beam sufficient to carry double the actual load if evenly distributed. For example, suppose the load to be carried is 4000 pounds, and the car itself weighs 2300 pounds. The total load is 6300 pounds. In estimating the proper size of beam, the load should

be considered as 12,600 pounds; if a wood crossbeam is to be used, care should be taken to select a sound straight-grained plank free from checks and knots.

*Double Wood Beam.* Some makers use two planks for the main crossbeam, as illustrated in Fig. 230. In such cases a space is left between them and their outer faces are set flush with the edges of the stiles, the top ends of which are cut away to form a tongue. This tongue passes between the beams, which are bolted to it. In such cases fillers of wood are fitted in between the beams at their top and bottom edges, being held in place by $\frac{5}{16}$- or $\frac{3}{8}$-inch bolts passing through both beams and fillers. This method of construction keeps the beam laterally rigid. As few bolts as possible should be used

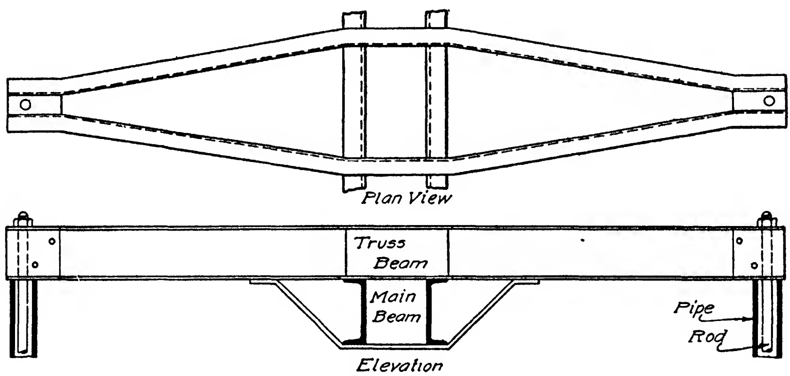



Fig. 231. Steel Truss Beam for Corner-Post Car

to keep the fillers in place, because every hole bored through the beams between the stiles tends to weaken them.

*Steel Beams.* When the main crossbeam is made of steel, channels are generally used, though some makers use two $\frac{1}{2}$-inch flat bars, varying in width from 6 to 10 inches as required. Cross-sections of these types are shown in Fig. 234. In estimating the size required, each beam should be amply strong to carry half the total load. In the case of the flat bars rigidity is obtained by the use of bolted spreaders between the bars. When channels are used no spreaders are needed, the flanges of the channel being sufficient to stay them laterally.

**Use of Truss Beam for Corner-Post Car.** In the case of corner-post cars there are no side braces used, for structural reasons, but the

load which is sustained at two opposite corners by the stiles is carried at the two other corners by means of what is known as a truss beam or crossbeam, Fig. 231. This beam is always made of iron or steel in several forms. Some makers prefer flat bar iron, while others prefer angle iron. However, two channels which are somewhat lighter than the main top beam, and which are bent to proper shape and fastened to the top beam by bolts through the flanges of each with a lower brace from the bottom of the main beam to the bottom of the truss beam, will be found as cheap and as effective a method as any.

Whatever design is used, the truss beam should be spread where it joins the main beam, so as to pass around the strap or lifting plate. The ends are brought together to allow long rods to pass through them down to and through the corners of the platform, as shown in Fig. 219. These rods are made of a diameter to suit the size of car and the load to be carried. A piece of steel pipe is cut to fit between the truss and platform, and the rod which is threaded at each end is fastened by nuts.

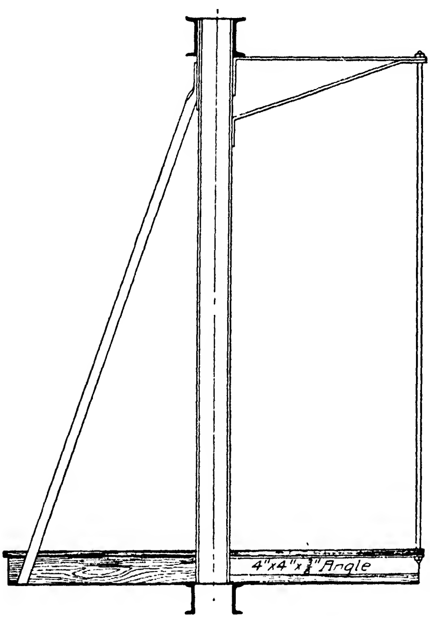


Fig. 232. Special Side Truss for Side-Post Car

A corner-post car at best is but a makeshift used only where entrances to the car are required on adjacent sides. The strains are different, the car being lifted from the corners, two of which are held fairly rigid by the stiles and guides. Any load that is placed on the floor of the car at the opposite corners has a tendency to rack or twist the main beam of the car through the medium of the corner rods and truss beam. Hence, the main beam should be made heavier than for a side-post car of the same span and weight in order to resist this twisting strain. As a result, corner-post cars weigh more than side-post cars of the same floor area.

**Special Truss for Side-Post Car.** In some cases all the entrances to a hatchway are from opposite sides except, perhaps, at one or two landings, and these one or two can be smaller. In such cases it is customary to use a side-post car, but at one place on the car the side brace is dispensed with and a special truss devised to make entrance possible there. Such an arrangement is shown in Fig. 232.

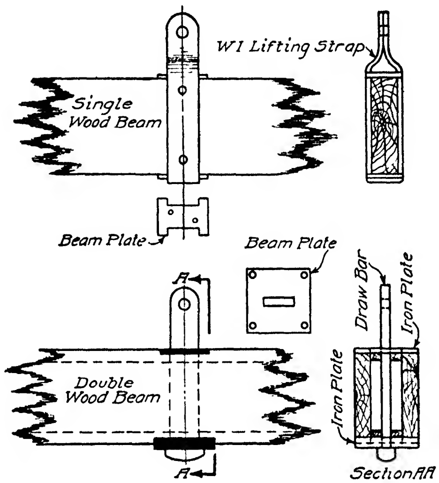


Fig. 233. Lifting Connections for Wood Beams

### Car Lifting Connections

**Lifting Straps.** With very few exceptions, cars are lifted at the center of the beam. In the case of a wood beam made of a single plank a strap is used, varying according to the size and the weight of car and

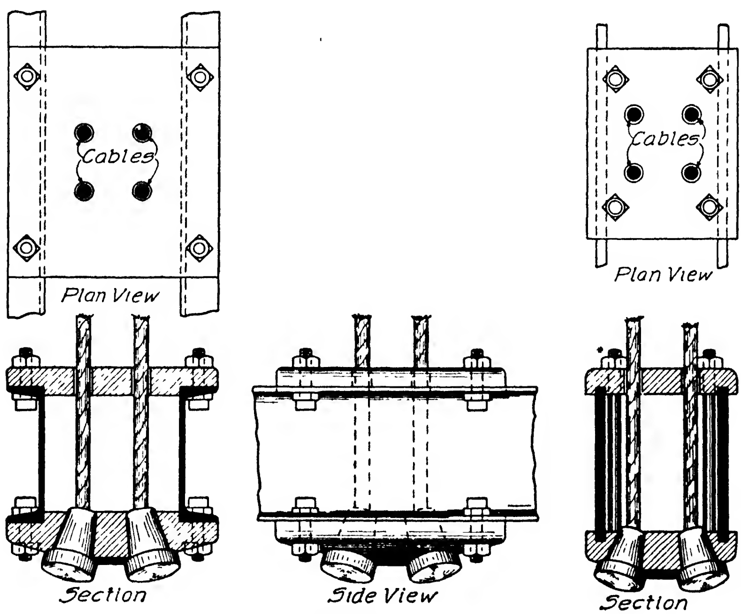


Fig. 234. Lifting Connections for Steel Beams

load from 2 by $\frac{1}{4}$ inches to $3\frac{1}{2}$ by $\frac{5}{8}$ inches. The form of this strap and the method of applying it to a wood beam is shown in Fig. 233. Beam plates keep the straps in place, distribute the pressure over a greater area, and prevent the wearing away of the beams.

**Drawbars.** Where there is a double wood beam a drawbar is used, which may be 2 by $\frac{1}{2}$ inches, $2\frac{1}{2}$ by $\frac{3}{4}$ inches, or 3 by $\frac{7}{8}$ inches, according to the requirements. The lower end of the drawbar is formed like the head of a bolt, and beam plates of iron are used at the top and bottom of beams, Fig. 233.

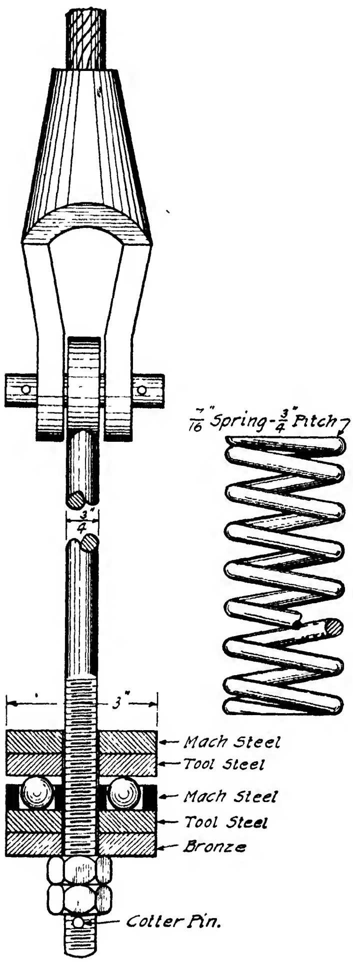


Fig. 235. Lifting Connection for Traction Elevator

**Lifting Plates.** With steel beams, whether of flat bar or of channel, the lower beam plate is of cast iron or cast steel, and from two to four holes are cored in it to receive the steel thimbles which form the cable fastenings, Fig. 234. These cored holes are set in such a position as to lead the cables toward the center, where they emerge through the top of the bottom plate. The cables are led straight up through the top plate, which serves simply as a guide for the cables, and which therefore is always made very much lighter and of cast iron. The lower plate is made very thick, in order to accommodate the thimbles and because the coring of such large holes for the thimbles weakens the plate. These beam plates are kept in position by four bolts. The bolts for the channel type of cross-beam pass through the flanges, and those for the flat-bar type pass through both plates and between the two bars which form the beam of the car, and simply clamp the plates in position. The

plates in this case are cast with grooves for the reception of the edges of the beams.

**Special High-Speed Connections.** With high-speed elevators, Fig. 254, it is customary to use special lifting plates of steel. Generally, a short section of channel riveted between the main beams of the car is used, the web of the channel being drilled for the reception of special drawbars, Fig. 235, fitted with alternate steel and bronze plates and steel balls. These permit the drawbars to revolve at will, thereby adjusting themselves to any twist the cables may acquire in running over traction drums or sheaves. The shock attendant on stopping and starting is lessened by the use of spiral steel springs on these drawbars between the revolving disks and the draw plate.

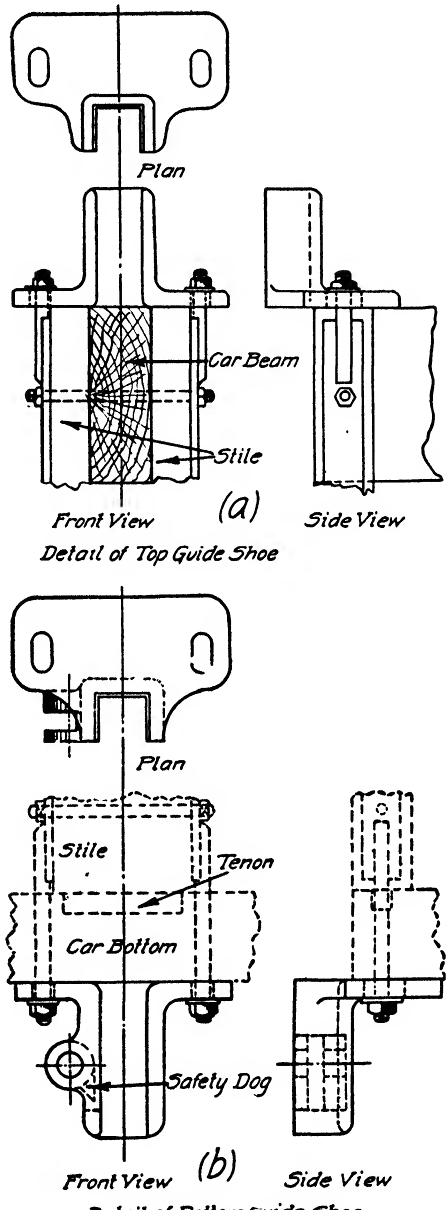



Fig. 236. Guide Shoes for Maple Guides

### Guide Shoes

**Shoe for Maple Guides.** The form of guide shoe in use with maple guides is shown in Fig. 236, which gives the details of both top and bottom shoes. They

are made with a flat base to be bolted either underneath the safety plank below the platform or on a plate on the top of the crossbeam of the car. That portion of the shoe which runs on the maple guide is milled or planed out smooth, true, and of proper size.

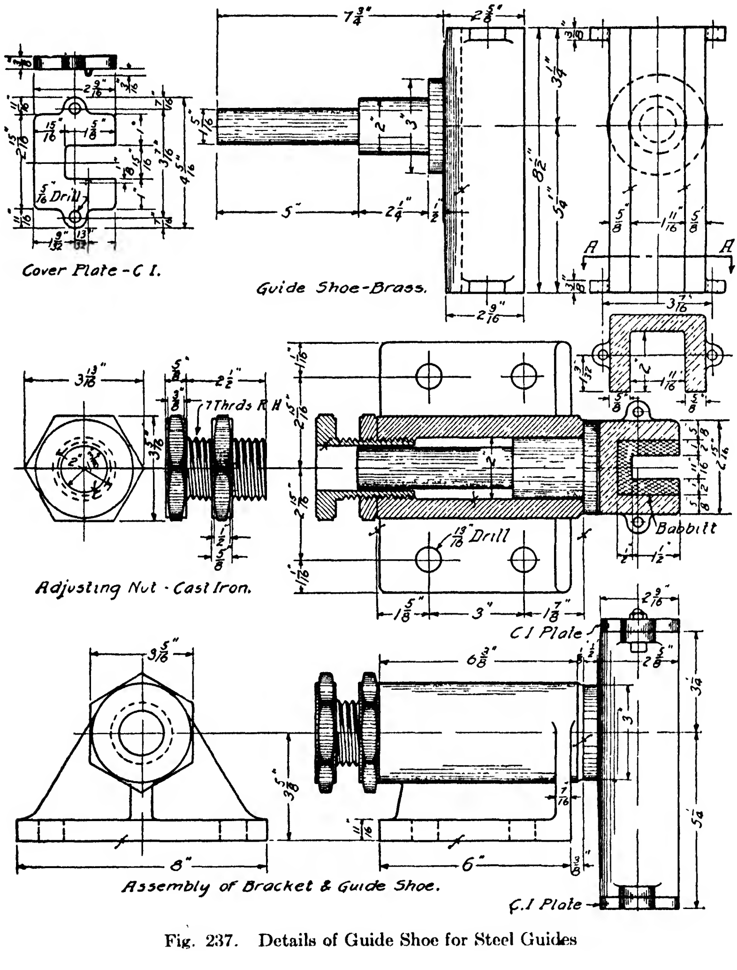



Fig. 237. Details of Guide Shoe for Steel Guides

*Setting.* Care must be taken in setting these shoes in place that the plates or plank on which they rest are square with the maple guides, otherwise they are liable to cramp the guide and bind the car. The smooth and perfect running of the cage is largely dependent upon the setting of these shoes.

**Shoe for Steel Guides.** *Construction.* For the steel guides a different form of shoe is used, Figs. 237 and 261. It differs slightly in that the shoe itself, which is frequently made of good hard bronze, is provided with a stem fitted into a socket or cylinder bored in the stand which sets on the safety board or beam. This method of construction permits the guide shoe to adjust itself. Hence it is not so essential that the base of the shoe should be so accurately square with the guide strip except at its face. At the back of this stand a set screw with a jamb nut is fitted, which is used to adjust or set out the shoe to gage, while a stout spiral steel spring or one made of rubber is usually inserted between the end of the set screw and the end of the stem of the shoe.

*Operation.* The effect of the compression spring is to hold the shoe always tight against the guide and prevent the bumping which is experienced frequently with the maple guide and shoe where they are slightly out of line. With this arrangement the shoes automatically adjust themselves to any slight inequalities of the guide, and with the steel guide these inequalities are reduced to the minimum. This form of guide, being unaffected by climatic changes, and being set up independent of the floors of the building, usually stays in good shape much longer than wood posts. Grease cups or automatic oilers of some other form are used with these guide shoes, for it is absolutely essential with a high speed and long run that this feature should never be neglected.

## COUNTERPOISING

### Early Forms

**Development.** In this section are described the construction and methods of application of the various forms of counterpoise weight used, from the time of the introduction of such weights up to the present.

**Nonadjustable Weights.** *Pocket Type.* The first counterpoise weights were long and of square section, being made at least 6 inches square and from 4 to 8 feet long. An eye of wrought iron was cast in one end by which they were attached to the cable and by which they were lifted. They did not run in guides but in a square tube or pocket extending the entire length of the run, Fig. 238. These weights were usually located at some distance from the hatch-

way. Such an arrangement would hardly be tolerated today. They were unwieldy and nonadjustable, and made a scraping noise when moving. Their only redeeming feature was their isolation from the hatchway, which is desirable in many respects, and is the present European practice.

*Flat Weight.* At the sides of the hatchway, where the guides are located, there is always considerable space both in front of and behind each guide. It was a desire to utilize this space which suggested the idea of a flat weight. The first of these weights were cast solid and on their longest edges were equipped with lugs designed to run in grooves plowed in soft pine timbers about 4 by 4 or 4 by 5 inches. The lugs were used as guides for the weights and were nonadjustable. Fig. 239 illustrates this weight, which was usually from 3 to 5 inches in thickness and of a width to suit the space in which it was to run.

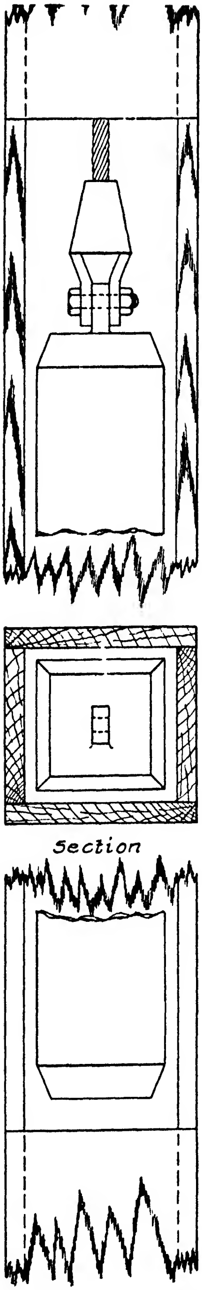


Fig 238 Pocket Form of Counterweight

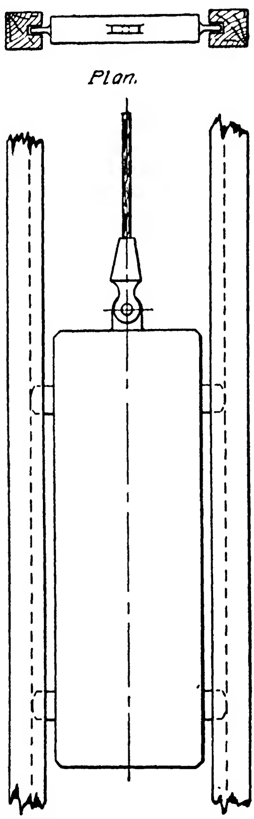


Fig. 239. Solid Flat Counterweight

**Adjustable Forms of Weight.** *Introduction of Frame Weights.* For many years after the introduction of the elevator, counterpoising was very crudely done. The method of doing this at that time—attaching the weight cable to the top of the car—did not permit a complete counter-

poise or balance, for if this were done the car would not descend. Hence, it was customary to make the weight anywhere from 500 to 800 pounds less than what the car itself weighed. This required more power to be used in lifting than if the car were fully counterpoised.

The introduction of machines of limited power in isolated plants where the elevator was driven by a small steam engine or a gas engine, caused elevator builders to see the advantage of over-counterpoising — that is, of making the weight heavier than the car, thereby helping the engine to lift the load. This could only be done by attaching the weight cable to the winding drum at the side opposite to that on which the lifting cable was fastened. This arrangement also led to the necessity of using an adjustable weight, the first form of which was the so-called frame weight, Fig. 240.

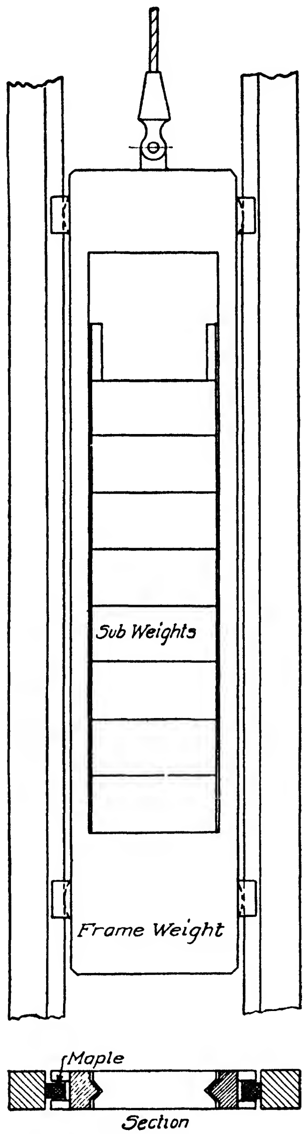


Fig. 240. Frame Type of Counterweight

The adjustment of the frame weight was made by the addition of a number of smaller weights which were made to fit over V-shaped guides. These guides were cast on the inside of the frame and terminated a sufficient distance below the top of the opening to admit the small weights. These small weights at that time were called subordinate weights, and now have the shorter name *subweights.* The grooved pine guides had not been found to wear well, so maple guides similar to those in use for the platforms were substituted, and the lugs were cast on the weight to suit this form of guide.

One disagreeable feature of the frame weight was its tendency to break in case of accident; a rather common occurrence

before the enactment of laws governing elevator installations. The subweights would scatter then in every direction, doing considerable damage if they fell from a great height.

*Use of Guide Weights.* This led to the adoption of top and bottom guide weights connected by wrought-iron rods, the subweights being grooved out at the ends to slide over the rods. When enough subweights were in the frame, either blocks of wood or short pieces of pipe were inserted and the whole bound together by tightening the nuts at each end of the rods. In other cases, pipes were used between the top and bottom weights, and the subweights were made to slip over the pipes. Such a weight is shown in Fig. 241. But even this form of weight was by no means perfect, for, in case of accidents whereby the weights became detached and dropped any distance, these rods and pipes would bend outward and permit the escape of the subweights.

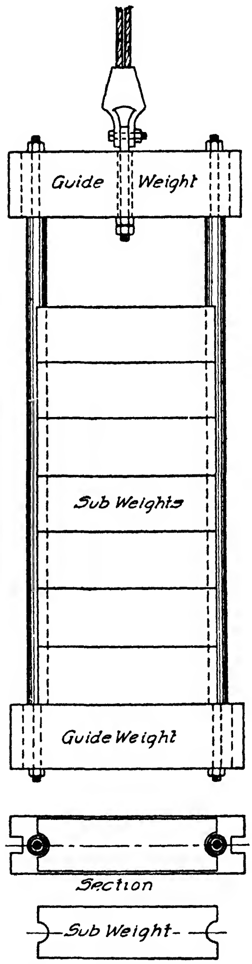


Fig. 241. Adjustable Counterweight Using Guide Weights

### Modern Counterpoise

**Development of Counterpoise Weight.** It finally became the custom to make the rods go through the subweights, casting holes in them for that purpose, so that even if the rods did bend the subweights could not escape. This is the counterpoise weight of today, Fig. 14 in Part I of Elevators, and Figs. 242 and 243, in general use on all types of drum machines for moderate heights and speeds.

*Drum and Car Weights.* The introduction of electricity as a motive power for elevators, and the ease with which undue friction and the excessive use of current could be readily detected by the use of the ammeter, brought about a nicety in the adjustment of the counterpoise weight which had not before been attained. It

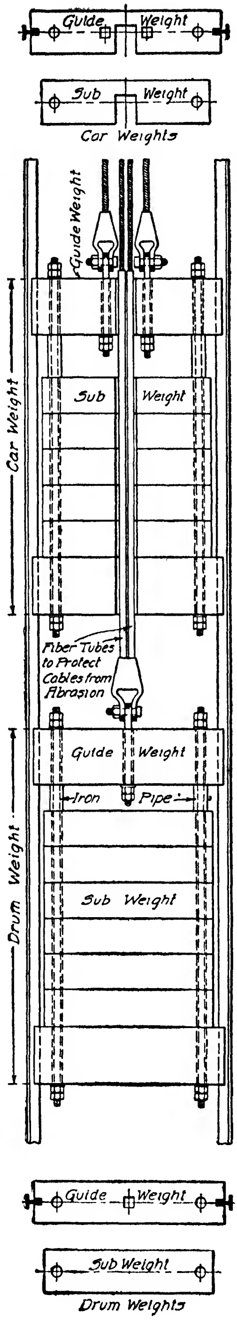



Fig. 242. Counterpoise Used with Engine in Basement

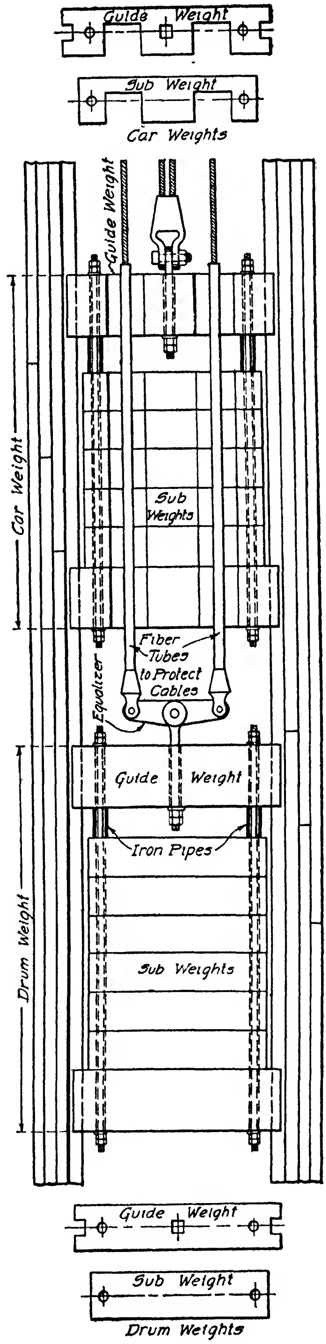



Fig. 243. Counterpoise Used with Engine Overhead

became the custom to counterpoise both from the car and from the drum on the same machine, as in Fig. 186 — the same installation as in Fig. 150, Electric Elevators, Part III — and Fig. 207, the counterpoise being attached to the car principally to relieve the lifting cables from undue strain. In Fig. 242 are shown in detail the weights arranged for attachment to an engine and car in the case where the engine is located on a foundation alongside the hatchway or on a floor of one of the lower stories. Fig. 243 is shown for the case where the engine with wide drum is at the top of the hatchway. The upper or car weights in both cases are made with slots or channels for the passage of the cables from the lower weight.

*Cable Passages.* As the drums are grooved differently in each case, the slots in the weights must be arranged differently. For an engine set beside the bottom of the hatchway, the grooves for the cables run parallel. This is not essential, but it is the usual way. They lead up to sheaves overhead and drop down in pairs to the weights below, and therefore only one groove is required in the car weight. Where the engine with a wide drum is located overhead the cables are led in grooves running singly from the center of the face of the drum out toward the ends, this being required because it is necessary in order to pull the car and weight centrally. In this case, two grooves — one for each cable — are required in the car weight, and they must be wider than in the former case, to allow for the travel of the cable on the drum. In all cases where the drum-weight cables pass through these slots or grooves in the car weight they must be protected from abrasion by shields made of tubes of hard fiber, as shown in the detail illustrations.

**Counterpoise Car Developed for Traction Elevator.** The advent of the modern skyscraper, and the consequent necessity for elevators of high speed, produced the traction electric elevator, Figs. 151, 161, 164, 165, and 168, Part III of Elevators, and Fig. 254, a machine in which the cables do not travel across the face of a wide drum but run constantly in the same grooves. The great speed and high travel of these machines developed the need for an improved form of counterpoise weight, and the steel frame, or counterweight car, is the result.

*Frame Construction.* This form of weight, Fig. 244, comprises a steel frame or car with sides of channel iron forming a sort of grooved

frame into which the subweights fit. It is provided with top and bottom beams similar to the cage which carries the car proper in which the goods and passengers are carried. However, it has no platform, the subweights being deposited within the frame itself and held there securely by two bolts passing through the bottom of this car. The frame is fitted with guide shoes, and these are adjustable, as are those on the car proper.

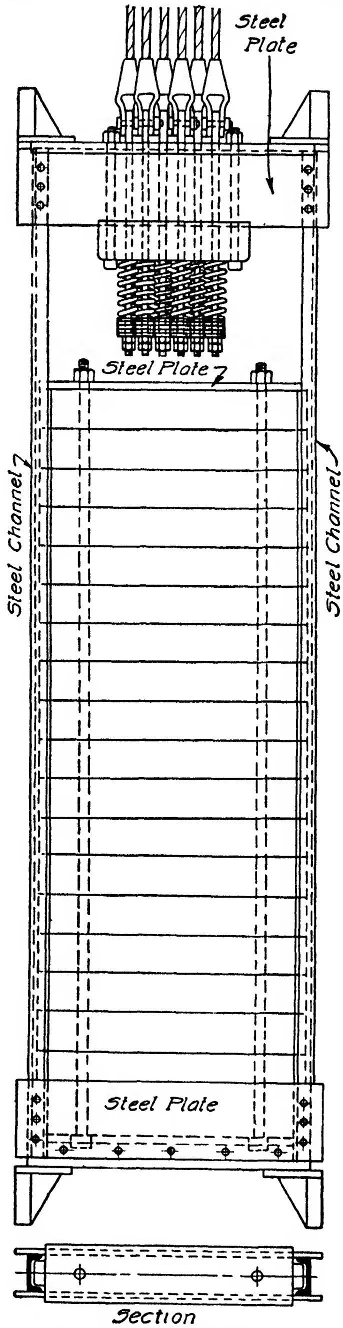


Fig. 244. Counterpoise Car for Traction Elevator

**Counterpoise Cable Attachment.** The cables, usually six in number, are attached to the weight frame by long drawbars or eye-bolts passing through suitable holes in the top beam. These extend clear through the beam and for some distance below it, where they are provided with steel springs. Below the springs is a series of washers or plates made alternately of steel and bronze. Double nuts are placed below these, and the ends of the drawbars are perforated for the insertion of cotter pins to prevent the possibility of the nuts coming off. The center plate of each series of bearings is perforated with 8 equidistant holes to receive 8 steel balls, each $\frac{1}{2}$ inch in diameter. All these plates are made $\frac{1}{2}$ inch thick, except the middle plate with the perforations, which is only $\frac{3}{8}$ inch thick. The effect of this arrangement is to cause the steel plates

above and below the center plate or retainer to bear directly on the balls.

*Automatic Action.* The conditions in this arrangement are as follows. In other types of machines in which one end of each cable is attached fixedly to the drum the cables simply wind on and off the drum without any tendency to twist, but in traction machines where one end of each cable is attached to the car and the other end to the weight — the lifting being done by the traction or adhesion of the cables to the grooves in the drum — there is frequently a rotative movement on the part of the cable as it enters or leaves the groove which is the result of the twist or lay of the rope in its manufacture. This is an undesirable feature because it causes the rope to lengthen or shorten slightly as it twists or untwists, and if not remedied, would cause the load to be divided unevenly between the various ropes or cables. The balls and plates used permit the rope which has lengthened or shortened to adjust itself automatically every time the elevator is stopped. The operation of this arrangement can readily be seen by anyone standing at one of the intermediate landings and watching these drawbars when the elevator stops.

As has been described in connection with Fig. 235, the ends of the cables attached to the car proper are fitted in exactly the same way as those attached to the weight. The springs are used to soften the stopping of the elevator, that is, to make a smooth easy stop free from shock or jar.

**Compensator Used with Counterpoise.** Under the heading of Hydraulic Elevators — Part II of Elevators — there is explained how the different positions of the car and counterpoise in the course of their travel affect the relative effectiveness of the counterpoise, and the method of regulating it by means of chains is mentioned. A later and more effective way of accomplishing this is by the compensator.

*Usual Cable Arrangement.* Cables equal in number and size to those used in lifting the car are attached to the car below and led beneath suitable sheaves placed in the bottom of the pit and thence up to the counterpoise weight, thus forming what is known as the compensator. Then, regardless of what the relative positions of the car and weight may be, the effectiveness of the later is undisturbed.

Fig. 245 is a diagram illustrating this arrangement. The sheaves are usually set in frames, which allow them to rise and fall a limited distance to accommodate any shortening or lengthening of the cables. To preserve a suitable tension, spiral steel springs are used

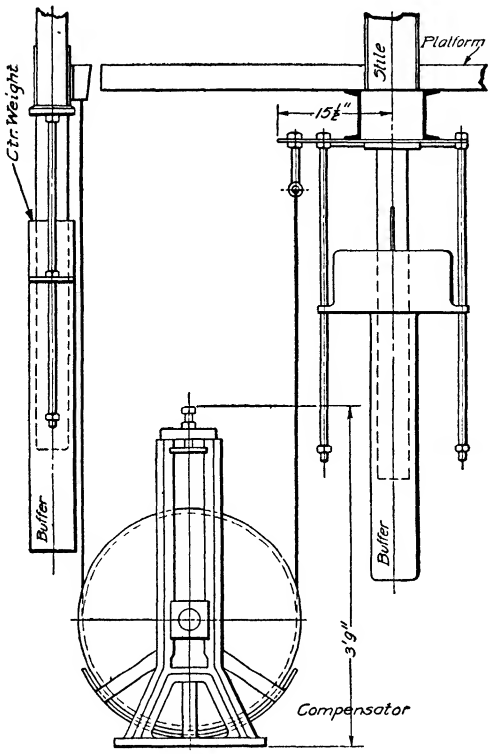
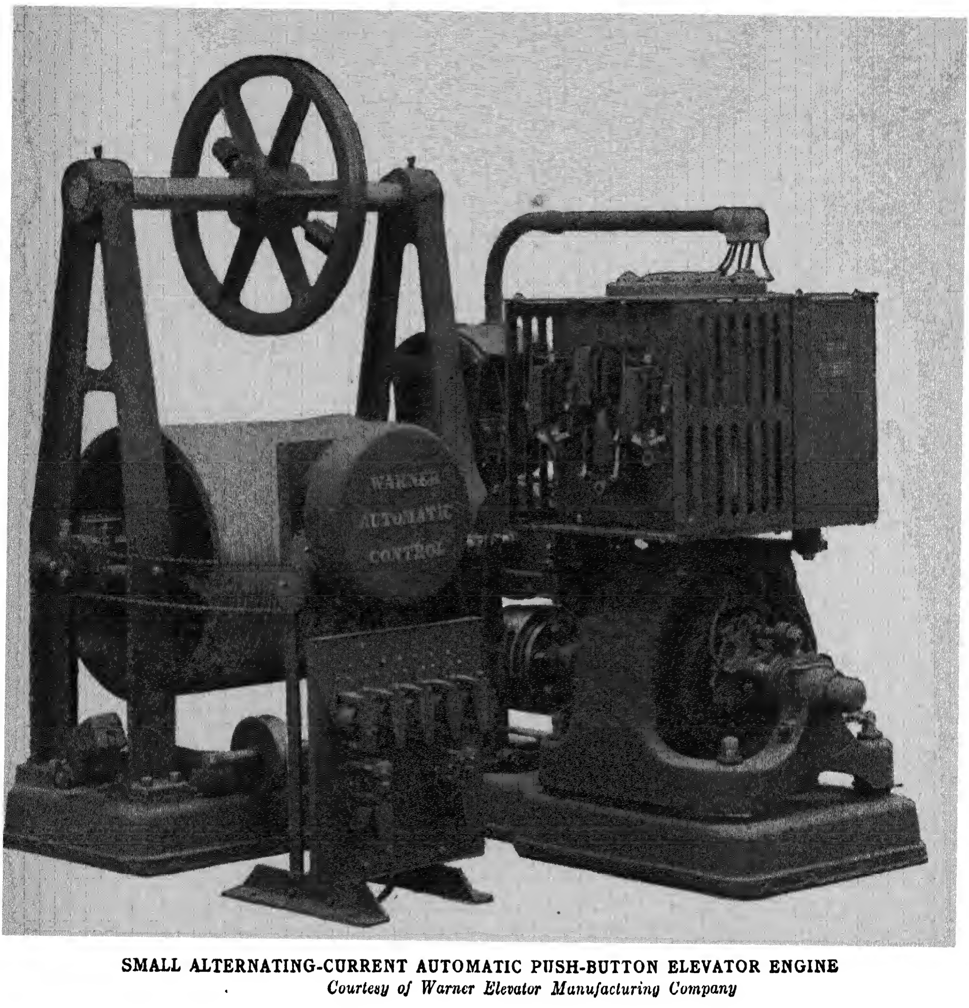



Fig. 245. Arrangement of Cable Compensator
*Courtesy of Kaestner & Hecht Company, Chicago*

in the frames above the boxes in which the journals of the sheaves run.

*Chain Compensator.* A cheaper way to accomplish this is to use one or more chains, equal in weight to the lifting cables and attached in the same manner, but dispensing with the sheaves. The chains are allowed to hang in a loop a little above the floor of the pit. The chains are likely to be noisy, and frequently when the elevator is stopped quickly they swing or sway in an unpleasant manner.




**SMALL ALTERNATING-CURRENT AUTOMATIC PUSH-BUTTON ELEVATOR ENGINE**
*Courtesy of Warner Elevator Manufacturing Company*

# ELEVATORS

## PART V

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## EQUIPMENT DESIGN AND CONSTRUCTION
(Continued)

### HYDRAULIC PARTS

#### CYLINDER

**Thickness of Walls.** *Rule.* To find $t$ the proper thickness of metal for a cast-iron cylinder, Fig. 246, multiply the bore $D$ of the cylinder in inches by the pressure $p$ in pounds per square inch, add to the result 1500 times the square root of the bore in inches, and point off four places to the left of the decimal point, which is equivalent to dividing by 10,000. The statement of this relation for the thickness of a cylinder wall is $t = \frac{pD + 1500\sqrt{D}}{10{,}000}$.

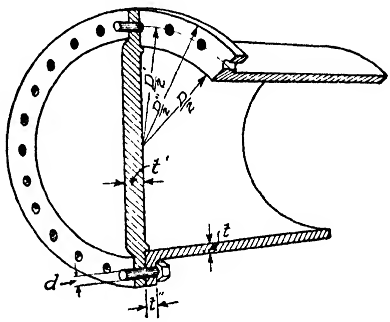


Fig. 246 Diagram of Hydraulic Cylinder Proportions

*Illustrative Examples.* *1.* For example, the bore of a cylinder is 30 inches, and the pressure 100 pounds to the square inch; what is the proper thickness? The product of 30 times 100 is 3000. The square root of 30 is 5.477, which multiplied by 1500 equals 8215. Adding 3000 to this result, 11,215 is obtained. By pointing off the fourth figure to the left for the decimal point, 1.12 inches or $1\frac{1}{8}$ inches is obtained as the proper thickness of metal to stand this pressure safely. Of course, if the cylinder is to be bored, a proper allowance of stock for this purpose must be added to this so that the thickness of metal when finished will be as figured.

*2.* As another example, try the same pressure of 100 pounds with a 6-inch pipe. Here 6 times 100 equals 600. The square root of 6 is 2.45 which multiplied by 1500 equals 3675. Adding 600 to

3675 gives 4275. Pointing off the fourth figure from the decimal point gives .4275, or in practice $\frac{7}{16}$ or $\frac{1}{2}$ inch.

**Size of Flanges.** *Thickness.* The thickness $t''$ of the flanges, Fig. 246, should be equal to the thickness $t$ of the metal in the body of the pipe or cylinder plus 40 per cent, allowing in addition to this the metal necessary for the joint and for truing up in the boring mill, and also for a fillet at its junction with the body of the cylinder; thus $t''=1.4t$.

*Diameter.* The flange diameter $D''$ should be wide enough for the bolts and nuts in addition to the cylinder outside diameter.

**Calculation of Flange Bolts.** *Total Tension.* The first step in determining the number and diameter of the bolts necessary to hold the cylinder head securely is to get the total load $W$, found by multiplying the area $A$ of the head in square inches by the unit pressure $p$; or the total stress $W=Ap$. For example, the area of a 30-inch cylinder is 706.86 square inches, which multiplied by 100 pounds per square inch gives 70,686 pounds as $W$ the total stress on the head to be carried by the bolts.

*Safe Tensile Stress.* The allowable unit tensile stress for wrought iron is 10,000 to 14,000 pounds per square inch, and for soft wrought Bessemer or open-hearth steel 12,000 to 16,000 pounds. Most bolts of today are made of the latter; hence it will be safe to take 14,000 pounds as our basis. The size of bolt, therefore, may be determined from the relation $N=\frac{W}{\frac{1}{2}s}$, in which $\frac{1}{2}s$ the allowable safe tensile stress per bolt, is $\frac{1}{2}$ of $14{,}000a'$, $a'$ being the minimum cross-section of the bolt. Either the diameter $d$, or $N$ the number of bolts, may be assumed, according to practice. Taking one of the sizes — $\frac{3}{4}$ or $\frac{7}{8}$ inch — customarily used for a 30-inch cylinder at 100 pounds pressure, it must be remembered that the effective diameter is at the root of the threads, and for a $\frac{7}{8}$-inch bolt is 0.73 inch, giving a net sectional area $a'$ of 0.42 square inch. This multiplied by 14,000 gives 5880 pounds as the limiting tensile stress $s$ on a $\frac{7}{8}$-inch bolt.

*Tightening Conditions.* It has been found in practice that under a pressure of 100 pounds to the square inch the bolts ought not to be farther apart than 7 inches, and that the best results are obtained at 4 or 5 inches between centers. This is due to the fact that when the bolts are farther apart than 5 inches it often becomes necessary to

tighten them to a greater degree than the allowable tension, in order to keep the joint tight under this pressure. To guard against this undue strain, the maximum tensile strength $s$ of the bolt is usually divided by 2 — reducing the safe allowable tension $\frac{1}{2}s$ to 2940 pounds on each bolt in this case. This safeguard is made all the more necessary from the fact that workmen frequently fail to tighten the bolts evenly. This does not occur with an experienced and careful workman, but it is liable to happen with a man of less experience, and in such a case the joint is very likely to leak on the side opposite to the one on which the bolts were tightened too hard in the beginning, in making up the joint.

The joint is usually made by using a rubber ring or gasket between the head and the end of the cylinder, these surfaces having been previously turned or faced true and flat. The gasket is sometimes made of lamp wicking, but it is harder to make in this form unless the workman is skilled in this particular way of doing the work; hence the rubber gasket is more popular. In any case, the bolts must be tightened evenly all around — slightly at first, screwing down the nuts with the hand, and afterward using a wrench, first on bolts at opposite ends of a diameter, then those at the ends of a diameter at right angles to the first, then upon bolts midway between those tightened, and so on — always tightening one bolt a very little, then the bolt directly opposite, and thus progressing until all bolts have equal stresses upon them. If this precaution is not taken, and the bolts on one side are tightened before those on the other, the gasket will be compressed on the side tightened first and the head will be slightly canted, and in this condition no amount of tightening of the other bolts will make a tight joint or one that will not leak under pressure.

*Rules for Number of Bolts.* *1.* Continuing the general example of a 30-inch cylinder and 100-pound pressure, with $1\frac{1}{8}$-inch walls and an allowance of a little more than half the diameter of the bolt on each side, the diameter of the bolt circle $D'$, Fig. 246, would be $33\frac{1}{2}$ inches and its circumference 105.2 inches. Suppose it is decided to set Bessemer-steel bolts $4\frac{1}{2}$ inches apart, then, simply dividing the circumference of 105.2 inches by $4\frac{1}{2}$, this spacing is found to be contained over 23 times, which would be an even number 24 in practice.

*2.* Let it now be seen how this works out from the equation $N=\frac{W}{\frac{1}{2}s}$, with the allowable stress $\frac{1}{2}s$ equaling 2940 pounds on one $\frac{7}{8}$-inch bolt. Dividing $W$ the pressure on the head, which is 70,686 pounds, by 2940, the resultant $N$ is 24. Hence, 24 $\frac{7}{8}$-inch bolts would be used.

The simple method of determining the number and size of bolts appropriate for any joint will be found quite reliable, the writer having used it with success for many years. Of course, the distance between the bolts is governed largely by the working pressure and by the thickness of the head, and also to some extent by experience. The exercise of a little judgment also is essential in the selection of the diameter of bolts to be used, as well as in deciding the distance between centers of bolt holes, but a judicious use of the rule just given will usually determine the latter point satisfactorily if the proper size of bolt is first selected.

## FEED VALVES

**Calculation of Valve Diameter.** *Rule.* To find the proper diameter of valve to feed the water into a cylinder fast enough to produce a desired speed of car under a given pressure and load, there may be used the following formula for $A'$ the area of the valve, which is $A'=\frac{L\times12\times A}{r\times60\times5\times\sqrt{h}}$. Here $L$ is car speed in feet per minute, $A$ is area of cylinder or piston in square inches, $r$ is ratio of gear reduction, and $h$ is pressure head in feet.

*Illustrative Example.* To make this clearer take an example as follows. What size valve is required for a car speed of 200 feet per minute, with a load of 3000 pounds, under a pressure of 100 pounds per square inch at the piston, the machine being geared 8 to 1?

The first thing to do is to find $A$ the area of the piston, which equals the load times the ratio of reduction plus the friction loss, all divided by the unit pressure; or $A=\frac{Wr+fWr}{p}$. By substitution, the piston area is 3000 times 8 plus .25 of 3000 times 8, all divided by 100; or $A$ equals 300 square inches.

The values may now be substituted in the equation for the valve area $A'$, remembering that 100 pounds pressure corresponds to a head

of about 230 feet. Thus, the area of the valve equals 200 times 12 times 300, all divided by the product of 8 times 60 times 5 times the square root of 230; or $A'$ equals 19.8 square inches. This is equivalent to a diameter of about 5 inches.

**Allowances for Obstructions.** *Two-Way Operating Valve.* Of course, in speaking of a 5-inch valve, it is intended to convey the idea of a valve that will deliver the amount of water that will pass through a clear pipe of the diameter named, and the valve should be so proportioned in all its parts that it will readily do this. In order to clearly understand what is meant, refer to the illustration of the ordinary two-way operating valve shown in Fig. 67 — Part II of Elevators. The water enters at the orifice marked "supply", and has a clear passage until it reaches the main portion of the valve, or what is called the barrel. Here the steel stem to which the plates holding the leather cups are attached passes down through the middle of the barrel, and, to the extent of its area, prevents the passage of the full amount of water that would go through if there were no obstruction.

*Valve-Stem Obstruction.* Let it be assumed that the valve is 4 inches, and that the stem is $1\frac{1}{4}$ inches in diameter. The area of a 4-inch tube is 12.57 square inches, that of a $1\frac{1}{4}$-inch stem is 1.23 square inches. It can be seen at once that the valve would give better service if its interior diameter were $4\frac{1}{4}$ inches, for its corresponding area, 14.18 square inches, minus 1.23 square inches would leave a net area of 12.95 square inches, or a little more than the area of a 4-inch pipe. Therefore the 4-inch valve really should be $4\frac{1}{4}$ inches in diameter to allow for the stem.

*Perforated Lining.* The next obstacle in the way of the free flow of water is the perforated lining of the valve where the water passes out to the cylinder. This perforated lining is made necessary in order to preserve the leather cups. It acts as a sort of cage, and at the same time as a guide to keep the leather packing in proper shape and to direct it properly in its motion from one point to another during the operation of the valve.

**Use of Leather Cups in Valves.** The leather cups at the lower end of the plunger at *a a*, Fig. 67, are those which control the passage of the water to and from the cylinder, while the leather cup at the upper end of the stem simply balances the valve so it will work easily and stay where it is placed for the time being.

*Relation to Balanced Pressure.* If the upper leather were omitted and the stem made to pass through a stuffing box at the upper end of the valve to prevent leakage of water, the entire pressure of water would be on the lower plunger, which if of 12 square inches area would be subjected to 1200 pounds pressure at 100 pounds to the square inch. It can be readily seen, therefore, how impracticable it would be to move the plunger against this pressure on one side of it, and also how impossible to keep it where placed. However, by having the upper disk or piston packed with the leather cup and attached to the stem and moving with it, the pressure against the lower disk or piston is balanced by the pressure against the upper one and consequently the plunger can be moved readily to any position and will stay just where placed.

*Relation to Valve Travel.* Since the stroke of the plunger is never sufficient to cause the leather cup to pass below the lower end of the brass lining, as shown in Fig. 67, the lining simply terminates there for the better flow of water through the valve.

The lower plunger has to be differently arranged. In the first place it is double — that is, it comprises two disks, each with its leather cup — and the distance between them should be sufficient to span the opening or passage to and from the cylinder, with a margin at either end so that it will not be difficult to center the valve when desiring to stop. It must be borne in mind that when the operator is standing on the floor of the moving cage, it is not so easy to center the plunger as it would be if he were on terra firma, and, if this margin is not provided, he will be very likely to move the plunger too far and thereby produce a reverse motion by opening the valve the other way. This would cause a very disagreeable jolt, and also cause the cables to jump or whip, which is bad for the machine. Therefore a margin of about $\frac{1}{2}$ inch is allowed at either end, that is, the part of the plunger which governs the ingress and egress of the water to and from the cylinder must be made fully 1 inch longer than the opening which it governs, and this must be in addition to the width of the lip on the leather cups.

**Disk-Valve Opening.** *Rule.* It is a rule in practice that if a common disk valve — such as a poppet, for example — is raised one-fourth of its diameter, it will allow the unretarded flow of all the water, or any other fluid or gas, which can pass through a pipe of

the same diameter as the valve. Therefore, as shown in Fig. 247, the full height of valve opening should be $h=\frac{d}{4}$.

*Illustrative Example.* To illustrate, taking the 4-inch valve as shown, the area of a pipe 4 inches in diameter is approximately $12\frac{1}{2}$ square inches, and the circumference is the same number of lineal inches. One-fourth of 4 inches is 1 inch, and, therefore, if a 4-inch valve is raised 1 inch, the opening is 1 inch times the circumference, or $12\frac{1}{2}$ square inches.

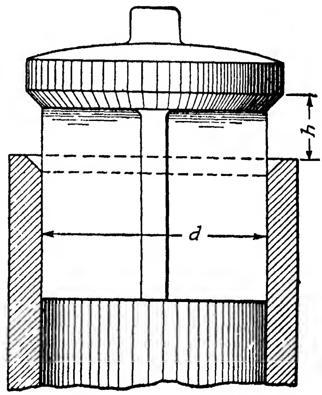

Fig. 247. Poppet-Valve Opening

**Perforated Linings in Two-Way Valves.** *Graduated Perforations.* In the case of the two-way hydraulic valve, Fig. 67, if the plunger is depressed or moved down 1 inch, the opening will be equal to the area of the valve, if there is a clear opening. But in this case the opening is partly obstructed by the brass lining, the only egress for the water being through the perforations, and these must be graduated in order to prevent a sudden starting up or down of the elevator, that is they must be smaller at one end than at the other. Accordingly, at each end of the space devoted to these perforations, the holes in the first row usually are made of $\frac{3}{16}$-inch diameter, the next row $\frac{1}{4}$-inch, then $\frac{5}{16}$-inch, and lastly of $\frac{3}{8}$-inch diameter for the middle three or four rows. Then they begin to decrease in size again to the other end of the space.

*Allowance for Perforation.* In determining the stroke of the valve, the area of all these holes or perforations has to be calculated, and, as the water in passing through them meets with considerable resistance, there must be enough holes in excess of 12 square inches to allow for the friction of the water in passing through them. This allowance should be from 50 per cent to 90 per cent, according to the size of the valve. Smaller valves require more allowance, because the holes cannot be made so large. In the larger sizes it has been found that a hole larger than $\frac{3}{8}$-inch diameter may not be used, because the lip or turned-over part of the leather cup cannot conveniently be

made more than $\frac{7}{16}$ inch to $\frac{1}{2}$ inch wide, and if it is less than the diameter of the hole it is apt to catch and cause the plunger to stick or damage the leather.

It will be found upon calculation that the plunger will have to move fully the diameter of the valve; in other words, the stroke of the plunger will have to equal the bore of the valve plus the lap or margin. Hence a 4-inch valve would have a stroke of 5 inches either way, or 10 inches in all. This stroke governs the length of the valve linings, the length of the sections of which the body of the valve is composed, and the distance of the upper section from the lower, as well as the distance apart of the ports or openings into and out of the valve. In designing a valve of this kind, the determining of the number of the perforations and the space they shall occupy is the second operation, the first being the inside diameter as before described.

*Construction of Lining.* The brass lining used is what is called seamless cold-drawn, and is usually $\frac{1}{8}$ inch thick. Being drawn through dies, it is as round, smooth, bright, and true as if turned and bored. The perforations are drilled after first being laid out or marked on the outside of the tube by means of a thin sheet-metal templet or pattern, which is wrapped around the tube. After the perforations are made and slightly rimmed or countersunk both inside and out to remove the arris or burr raised by the drill, the valve body is bored in the lathe to a rather tight fit. The brass lining is then forced into the body of the valve, care being taken not to use too much pressure — otherwise the brass lining will buckle or crumple, which will spoil it, and in that case there would be no remedy but to remove it and make another. The brass tube before being forced into place is smeared over with a thin coating of white lead in oil, partly to serve as a lubricant in the process of insertion, and also to act as a cement and insure a tight connection.

In order to assist in giving the water a free clear passage, that portion of the valve body which surrounds the perforations in the lining is enlarged so as to form an annular passage around the lining. This can be readily seen by reference to Fig. 67, in Part II of Elevators. The size of this passage must be determined by calculation, so as to guard against its being too small, and all these points must be carefully considered, if the valve is to be successful in meeting the requirements.

## CYLINDER HEADS

**Thickness of Plain Head.** *Rule.* To find the proper thickness $t'$, Fig. 246, for a plain cast-iron cylinder head, when not stayed or strengthened by ribs, multiply half the diameter $D'$ of the head between centers of bolt holes by the square root of the unit pressure $p$ in pounds, and divide the product by 71; thus, the thickness $t' = \frac{\frac{1}{2}D'\sqrt{p}}{71}$.

*Illustrative Example.* Take the cylinder previously used as an illustration; the diameter is 30 inches, and the pressure is 100 pounds to the square inch. The thickness of metal in the cylinder is $1\frac{1}{8}$ inch, and $\frac{5}{8}$ inch is allowed outside this on each side for the distance to the center of both holes. This makes in all $33\frac{1}{2}$ inches, one-half of which is 16.75 inches. The square root of the pressure, which is 100 pounds per square inch, is 10. Multiplying 16.75 by 10 gives 167.5, which divided by 71 gives 2.36 inches, or about $2\frac{3}{8}$ inches for the thickness of this plain head.

**Tapered Head Construction.** It must be remembered that the thickness just computed is for a plain head not stayed or ribbed in any way; but a head of this thickness would be not only unwieldy but would look somewhat disproportionate. It does not need to be as thick at the outer portion or rim where it bolts to the cylinder flange, because its weakest part is at the center. Hence, it could be made $2\frac{1}{2}$ inches thick at the center, and gradually tapered to $1\frac{1}{16}$ inches at the part covering the flange of the cylinder. The dimension $1\frac{1}{16}$ inches is obtained by taking the thickness of the metal in the sides of the cylinder, which is $1\frac{1}{8}$ or 1.125 inches, and adding to it 40 per cent of itself, as explained in the preceding section on the Hydraulic Cylinder in describing the method of finding the thickness of the flange.

**Ribbed Head Design.** A better way would be to rib the head; this could be done by making a boss, or hub, in the center of the head, and from this boss running ribs radiating toward the outer portion where the bolts are to be, as shown in Fig. 248. This would make an infinitely lighter head, and if proportioned properly would be just as strong as the plain head; but, because of the many different ways in which the ribs might be arranged, and their different thicknesses and depths, no set rule can be laid down for determining the thick-

ness of the head in a case like this. The only way to proceed is by an experimental calculation. Each rib must be taken as a beam, or, in other words, considered as a beam fastened at the ends and with an evenly distributed load on it.

*Illustrative Example.* Suppose, for example, that a rib 1 inch thick and 5 inches deep at the center and 1 inch deep at the ends, is 30 inches long. What load will it sustain safely?

As has been stated in the section on the Proportioning of Gearing the safe resisting power of cast iron when under transverse stress may be taken as 5544 pounds per square inch of transverse section.

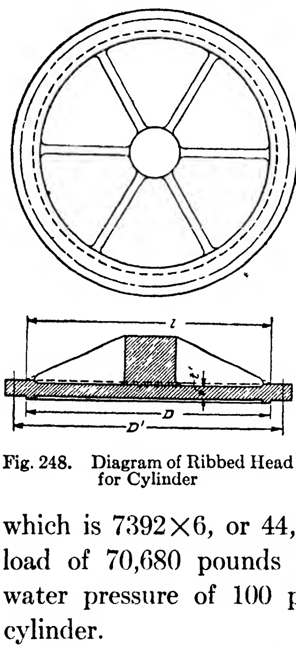

Fig. 248. Diagram of Ribbed Head for Cylinder

Let $s$ equal this safe modulus of rupture of 5544 pounds per square inch; $A$ the section of the rib in square inches; and $l$ the span in feet. Then the safe load in pounds on a cast-iron beam may be stated as $W=\frac{2A}{3l}\times s$.

The section of the rib at the center where the greatest stress exists is 5 square inches; the span, 2.5 feet; and the safe modulus of rupture, 5544. Hence, the safe load for this rib is $\frac{2\times 5}{3\times 2.5}\times 5544$, or 7392 pounds. But there are six ribs, the total strength or power resistance of which is $7392\times 6$, or 44,350 pounds. This is about $\frac{5}{8}$ of the total load of 70,680 pounds on the head resulting from the maximum water pressure of 100 pounds per square inch inside the 30-inch cylinder.

To take care of the remaining $\frac{3}{8}$ of the total pressure, 26,330 pounds, the thickness of the wall of the cylinder head may be calculated by the rule for plain heads, $t'=\frac{\frac{1}{2}D'\sqrt{p}}{71}$. While only $\frac{3}{8}$ of the total pressure is in excess of the load which the ribs may carry, suppose we proportion the plain part of the head to carry twice that, or about 65 pounds per square inch of area, making an allowance on the side of safety. Then, the thickness of plain head to carry that proportion of the pressure would be the product of $\frac{1}{2}$ times $33\frac{1}{2}$, or

16.75, and the square root of 65 — say 8 — giving 134, which, divided by 71, results in 1.88. Thus it would be safe to make the thickness of the head $1\frac{7}{8}$ inches in addition to the depth of the ribs.

This example shows the advantage of ribs used with judgment and discretion for the purpose of strengthening a casting. This form of head can be used very advantageously in cases where a piston rod passes through it, the boss in the center being cored out to serve as the stuffing box. Of course, care must be taken to have the sides as thick as the ribs, not forgetting to make provision for the studs necessary for tightening the gland on the packing.

But it so happens that in a thrust machine — that is, in a hydraulic cylinder where the pressure of the water on the piston pushes the piston rod outward — the end of the cylinder through which the rod passes is open and neither head nor stuffing box is used, as it is only on a pull machine that those parts are needed. The fixed or immovable sheaves at the rear end of the push machine are set in arms which are either cast on or bolted to the end of the cylinder, Figs. 79 and 87, Elevators, Part II, so that in a measure they counteract the pressure inside. To remedy this one or two strong ribs may be cast on the head between the arms, care being taken to provide for the proper clearance of the sheaves which are to go between the arms.

## PISTON RODS

### Pull-Machine Type

**Piston and Rod.** In the case of a pull machine — Figs. 77 and 86, Hydraulic Elevators, Part II — the piston rod is in tension when working, and the proper diameter is easily found by making the area of the rod sufficient to safely resist the tensile stress; but, in case the cylinder is long, the tendency of the rod to sag in the middle must be taken into account, and the rod made sufficiently large in diameter to support itself transversely. The rod is usually attached to the piston by fitting the rod into a tapered hole in the piston and putting a nut on the small end of the rod to keep it from pulling out, Fig. 249. The calculation for strength of rod to resist the pull must be based on the diameter at the bottom of the thread — as has been described in the calculation of cylinder flange bolts. The tapered hole, and the consequent fitting of the end of the rod to it, serves the

purpose of making a water-tight joint to prevent leakage of water through that part.

**Crosshead.** In the crosshead is bored a straight hole, and the rod is kept in place either by a nut or by means of a cotter. In either case, the section of the nut or cotter, and also of the rod on

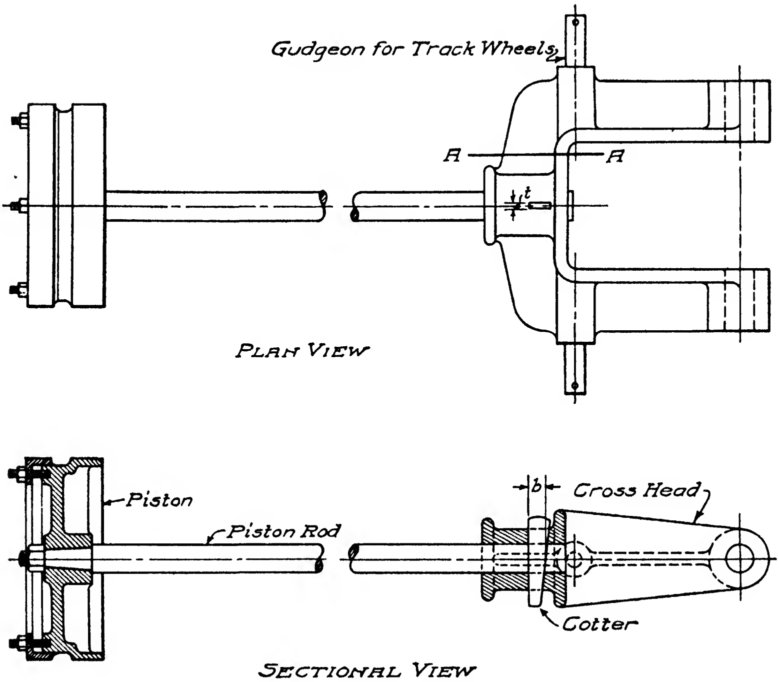


Fig. 249. Details of Pull-Machine Piston, Rod, and Crosshead

either side of the cotter, must be carefully figured to see that proper strength is provided.

*Rules for Cotter.* In calculating the size of the cotter, Fig. 249, use the rule for shear given under the section on Gearing for finding the dimensions of a key, remembering that the shearing stress acts at two places on the cotter and that, consequently, the mean dimensions of each of the sections in shear may be allowed for $\frac{1}{2}$ the area figured for a key.

By another rule for getting the proportions of a cotter through a piston rod — based on the rod being of proper diameter — $b = 1.463d$,

and $t=\frac{d}{5}$, in which $b$ is the mean breadth of the cotter, $t$ its thickness, and $d$ is the diameter of the piston rod.

### Thrust-Machine Piston Rod

**Rod and Piston.** In a thrust machine — Figs. 79, 85, and 87, Hydraulic Elevators, Part II — the piston rod is not turned in the lathe, there being no need for it, and it is usually made from a piece of

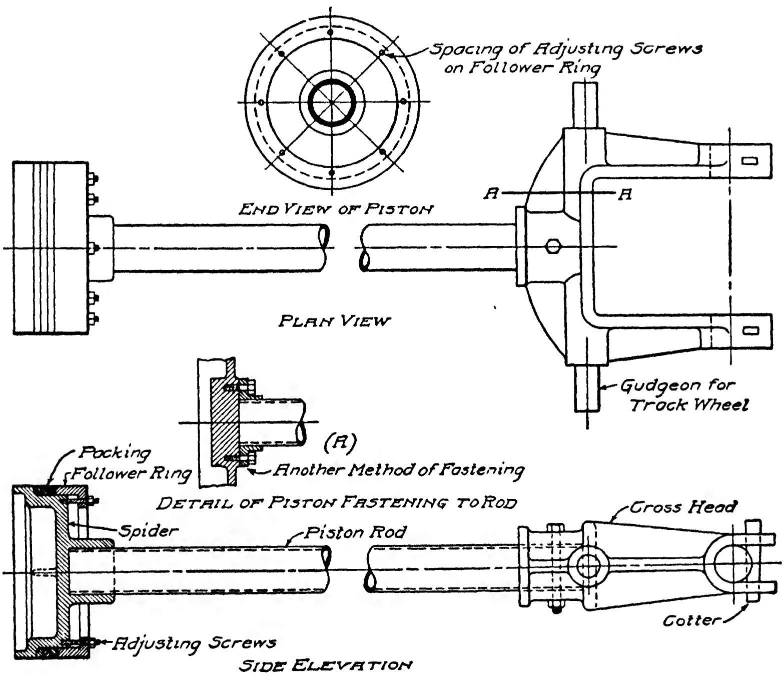


Fig 250. Details of Push-Machine Piston, Rod, and Crosshead

double-thick steel pipe. A proper socket or hollow boss is provided for the reception of the ends of the pipe on both the piston and the crosshead, Fig. 250, that on the piston being bored out while the piston is in the lathe for turning and fitting the follower thereto. The end of the pipe rod is turned just sufficiently to fit into the boss.

*Flanged Connection.* Some makers cut a thread on the end of the pipe and screw an extra heavy flange on tight, Fig. 80, Part II of

Elevators, the flange being faced off while the rod is still in the lathe, and afterwards is attached to the piston by bolts tapped into the latter, as seen in detail (*A*), Fig. 250. In this case the piston is made without a hub, a raised patch being cast on the piston of proper diameter and thickness which is faced off while the piston is in the lathe. A similar patch is cast on the opposite side to provide enough metal for tapping without drilling through the piston; this is to prevent leakage.

**Crosshead.** In fitting the piston rod into the crosshead it may be done by leaving sufficient room around the pipe inside the hub for the reception of a lining of some easily fused metal such as babbitt. After lining up the cylinder and ways, the piston, crosshead, and rod may be put in place, lined up, and the metal poured around the rod where it is placed in the hub of the crosshead, care being taken to plug the end of the pipe with a disk of wood to prevent the melted metal from flowing into the pipe. Afterwards a bolt is put through both pipe and hub to prevent any possibility of the two members pulling apart.

That part of the crosshead most liable to breakage is through the line *A*—*A* in Figs. 249 and 250. The method of determining the proper amount of metal and the best disposition of it to resist the stress to which it is subjected is found by the application of the rule given for finding the proper strength of the arms of a gear. The arms of the crosshead, being under compression in the case of the push machine and under tension in the pull machine, must be figured differently in each case, but at *A*—*A* the stress may be considered the same in both cases.

Referring to Figs. 249 and 250 showing the crosshead and piston of the two kinds of machines under discussion, the points just described will be more easily understood. Attention is directed to the ribs on the back of each piston. They are a further illustration of the method of lightening the casting by using strengthening ribs.

## MISCELLANEOUS CONSIDERATIONS

**Gudgeons and Track Rails.** The gudgeons in this connection are axles for truck wheels, which run on a track usually consisting of two 6- or 8-inch I-beams, which rest on legs attached at their lower ends to piers built in the ground for their proper support.

The truck wheels should be 10 or 12 inches in diameter at the tread, and should be flanged to keep them on the track. Usually $1\frac{15}{16}$ inches is the proper diameter for the gudgeons, but this may be

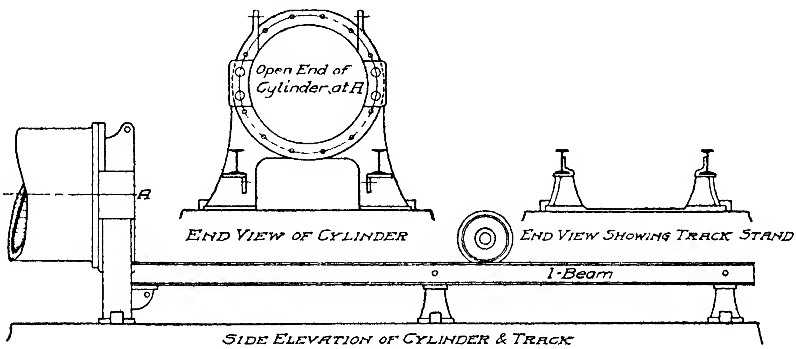


Fig. 251. Cylinder and Track Supports for Horizontal Hydraulic Machine

determined easily by one of the rules for shaft diameter already given. Fig. 251 shows the track and stands very clearly, as well as the saddles on which the cylinder sets.

**Piston Lubricating Device.** It is well to describe here a device which is frequently used on these machines for the purpose of keeping the interior of the cylinder lubricated — this being a very important feature in the life and serviceability of the machine. Fig. 252 shows a section of the rim of the piston *P* with the follower ring *F* and packing. In this case the follower ring is made longer than is shown in the preceding sketches, to make room for the annular space *A* which is a

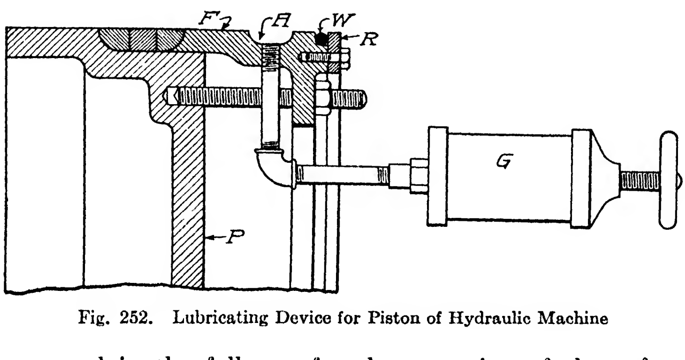


Fig. 252. Lubricating Device for Piston of Hydraulic Machine

groove turned in the follower for the reception of the soft yellow grease sold for the purpose. To supply this space with grease, a pipe is tapped into the ring — not less than a $\frac{3}{4}$-inch pipe is used —

### TABLE IV

Safe Fiber Stresses

| MATERIAL | TRANSVERSE LOAD (lb. per sq. in.) | LOAD IN TENSION (lb. per sq. in.) |
|---|---|---|
| Cast iron | 5,544 | 3,000 |
| Wrought iron | 12,000 | 10-14,000 |
| Steel | 16,000 | 12-18,000 |

and a compression grease cup is attached as shown at *G*. As the annular space is kept full of grease, it lubricates the cylinder bore at every stroke of the piston. To prevent waste, there is used a wiper, *W*, which is simply a ring of about $\frac{1}{2}$- or $\frac{5}{8}$-inch pure rubber cord held in place by the outside ring *R*. On the return stroke of the piston this rubber ring wipes the superfluous grease back again into the annular space. The compression grease cup should be at least 3 or 4 inches in diameter and 10 inches long, in order to hold a sufficient supply. By the use of a long pipe it may be located at the crosshead if desired.

## Strength of Materials

**Safe Fiber Stresses.** In those hydraulic elevators operating under a high pressure of from 300 to 800 pounds per square inch, cast iron is eliminated as far as possible. The plungers, and frequently the cylinders, are made of it, but the crossheads and all parts subject to severe transverse or tensile stresses are made of steel as far as possible; the same rules apply in finding their proportions. For the convenience of the student the safe fiber stresses for the materials entering into the composition of these machines are given in Table IV.

*Affecting Conditions.* The figures for transverse stress are based on $\frac{1}{4}$ of the breaking loads for steel and wrought iron and $\frac{1}{6}$ for cast iron. The variation of from 10,000 to 14,000 pounds per square inch in the allowable tensile stress for wrought iron is due to the various qualities of this material and the manner in which it has been worked in the production of the members under consideration. Much bending and welding necessitate the use of the lower figure. The use of the common grades also requires this precaution, the refined irons being the more reliable unless spoiled in the forging or

other process. With steel the difference is due to the quality used, the mild steel being not quite so strong as the harder qualities.

## POWER REQUIREMENTS

**Definition of Power.** *Power* is a term used to express the time rate of doing mechanical work — that is, the capacity to exert a certain force for a certain distance during a unit of time.

*Unit Rate of Work Fixed by Watt.* Before the introduction of the steam engine there was no standard unit of power. Power at that time was obtained either from the exertions of animals or from machines driven by wind or water. When James Watt introduced his steam engine to drive mills and other machinery, about the year 1780, he found it necessary to obtain some unit by which the power of his engines could be expressed. This he decided to do by comparing the performance of his engines with that of a good horse. He therefore made a number of tests with the heavy horses in use at that time in the breweries in London, and found that the strongest ones were able to work at a rate equivalent to raising 33,000 pounds through a vertical distance of 1 foot in the space of 1 minute. This rate of performing work was called the horsepower.

*Rule for Horsepower.* A *horsepower* is said to be equal to 33,000 foot pounds per minute — the unit termed *foot pound per minute* representing the product of a force of 1 pound exerted through a distance of 1 foot for 1 minute of time. Thus when $W$ is the force in pounds acting through a distance $L$ in feet in 1 minute's time, the amount of horsepower $P$ may be determined by the relation $P=\frac{WL}{33,000}$.

### POWER FOR HAND OPERATED ELEVATORS

**Extent of Human Exertion.** In those elevators operated by manual labor the term horsepower is not used. It has been found by actual test that, while a man is capable of exerting considerable force during short periods of time, the average force exerted by a laborer either in turning a crank or pulling on a rope does not exceed 30 pounds. Hence, either in estimating the force required by manual labor for lifting a load, or in calculating the gearing between the operator and the load, this is the standard adopted.

**Calculation of Loads.** *One-Man Capacity.* Take, for example, the case of a hand elevator where the ratio between the hand rope and the load is about 25 to 30. A man, therefore, pulling on the hand rope with an average force of 30 pounds could raise a load on the platform of about 700 to 800 pounds, allowing 15 per cent for friction. Elevator builders aim to make one man's labor raise about this amount in designing hand-power elevators. Of course, for a short period it is quite possible for one man to raise three times that amount of load, but not without great exertion.

*Power Required for Heavy Loads.* On hand elevators built to lift from 2000 to 3000 pounds, it is intended that 3 to 4 men will be used in lifting the full load. These elevators are usually so arranged that two men can pull on the hand rope at each floor, Figs. 10, 13, and 20, Elevators, Part I.

In this manner it is surprising what a large quantity of goods can be handled in a comparatively short time. Of course, to enable manual labor to attain such good results, every possible means of reducing friction is employed, and counterbalance weights heavier than the car are often employed as a means of assisting the elevation of the load.

## POWER FOR BELT DRIVEN ELEVATORS

**Effect of Counterbalance Weight.** In computing the horsepower for a belt driven elevator, that is, driven from a line shaft or countershaft, Figs. 37, and 40 to 42, in Part I of Elevators, it is not customary to consider the feature of over-counterbalance — because the counterbalance weight usually is hung from the rear side of the drum and it pulls against the hoisting cable, assisting it in lifting the load — but the counterbalance is seldom made heavier than the cage or platform. At other times the counterbalance cable is attached directly to the crossbeam of the cage and is led up over sheaves and down to the weight. In such a case the counterpoise weight must weigh considerably less than the cage, otherwise the cage would not descend.

**Calculation of Horsepower Required.** The horsepower required to operate the average 2-belt elevator driven from a line shaft or countershaft depends largely upon the design of the winding apparatus. Ordinarily the manufacturer uses as a basis of calculation two or three sizes of machine, the capacities of which are known beforehand from practical tests made with a view of determining them.

*Illustrative Example.* Take, for example, a machine of a capacity limited to 2500 pounds, the friction of which amounts to 33 per cent. Let it first be desired to lift 2000 pounds at a speed of 50 feet per minute. Multiplying 2000 pounds by 50 feet per minute, it is found that work will have to be done at the rate of 100,000 foot pounds per minute. Since 33,000 foot pounds per minute equal one horsepower, there would be required $\frac{100,000}{33,000}$, or 3.33 horsepower, if there were no friction. Since 33 per cent friction was assumed, 33 per cent of 3.33 horsepower, or 1.11 horsepower, must be added to the result, giving 4.44 horsepower as the total requirement.

*Formula.* This example illustrates the practical method of determining the power required to lift a given load at a definite speed on one of these machines, which for precision in the solution of such problems may be stated as follows: *horsepower required* is $P = \frac{WL}{33,000} + P_f$; in which $W$ is the load in pounds, $L$ is the speed in feet per minute, and $P_f$ is the per cent of horsepower lost in friction.

**Calculation of Belt Size.** To find the width of belt necessary to transmit this amount of power, the gear ratios and the diameters and speeds of the pulleys have to be considered.

*Speed.* Let it be assumed that the drum upon which the cable winds is 26 inches in diameter, or approximately 82 inches in circumference, and that the speed of the car is to be 50 feet per minute, or 600 inches per minute. Dividing the linear velocity by the circumference of the drum gives $\frac{600}{82}$, or 7.32 revolutions per minute as the speed of the drum. Multiplying this result by the number of teeth in the gear for a single-pitch worm gives the revolutions of the pulleys. Assuming the gear to have 50 teeth, the speed of the pulleys will be 7.32 times 50, or 366 revolutions per minute. The proper diameter of the pulley in this case would be 18 inches. Such a pulley would have a circumference of 56.5 inches, thus giving a belt speed of 56.5 times 366, equal to 20,679 inches per minute, or about 1723 feet per minute. If spur gearing were used, the speed of the drum would have to be multiplied by the gear ratio, that is, the number of teeth in the gear divided by the number of teeth in the pinion, as discussed in connection with Fig. 177.

*Application of Empirical Rule.* A rule much used among engineers is that every inch in width of a double belt is capable of trans-

mitting 1 horsepower at a speed of 800 feet per minute, 2 horsepower at a speed of 1600 feet per minute, and so on. Applying this rule to the present case shows that a double belt about 2 inches wide would do the work, provided the pulleys were of such size as to prevent slippage.

In practice, however, it is often found advisable to be liberal in the application of this rule. Sometimes the driving pulley on the line shaft is so much greater in diameter than the pulley on the machine that the arc of contact on the latter is less than it should be to provide the necessary adhesion between the pulley and the belt. Belts are often slack, and curl at the edges, so that their full width is not in contact with the face of the pulley. Taking these facts into consideration it would be advisable in this case to use a belt of not less than 3-inch width. A 4-inch belt is still better, for, if the gear and worm are of adequate strength, the machine mentioned will readily lift 3000 pounds, the gear and worm being of $1\frac{1}{4}$-inch pitch.

**Factors in Belt Selection.** The width of belt needed depends on three factors: (1) tension of belt; (2) size of smaller pulley, and proportion of surface touched by belt; and (3) speed of belt.

**Transmission Stress in Belt.** *Safe Allowable Load.* The average stress under which a leather belt will break has been found by many experiments with various good tannages to be 3200 pounds per square inch of cross-section, while a very good quality of leather will sustain a somewhat greater strain. When used on pulleys, belts should not be subjected to a greater stress than $\frac{1}{11}$ of their tensile strength, or about 290 pounds to the square inch of cross-section. This, for a single belt $\frac{3}{16}$ inch thick, amounts to 55 pounds average stress for every inch in width. The stress allowed for all widths of belting — single, light-double, and heavy-double — is in direct proportion to the thickness of the belt. If a greater stress is attempted, the belt is likely to be overworked, in which case the result will be an undue amount of stretching and damage to the joints or laps, or, in case it is a laced belt, the tearing out of the lace or hook holes.

*Mean Tensile Stress.* When the belt is in motion, the stress on the working part will be greater than on the slack part, for in every belt one of the spans stretching between the pulleys is doing the work while the other is under less stress. The average tensile stress $T$

in the belt is $\frac{1}{2}$ the aggregate of both sides, or $T=\frac{w+w'}{2}$, when $w$ and $w'$ are the tensions in the two opposite parts of the belt.

**Calculation of Belt Width.** *Belt Adhesion in Power Transmission.* The working adhesion of a belt to the pulley is in proportion both to the number of square inches of belt in contact with the surface of the smaller pulley and also to the arc of the circumference of the pulley touched by the belt. This adhesion forms the basis of all calculations in ascertaining the width of belt necessary to transmit a given amount of power.

A single belt $\frac{3}{16}$ inch thick subjected to a stress of 55 pounds per inch of width, when touching $\frac{1}{2}$ of the circumference of an iron pulley, will adhere to it with a force of $\frac{1}{2}$ pound per square inch of contact surface. If the belt touches only $\frac{1}{4}$ of the circumference of the pulley, the adhesion is only $\frac{1}{4}$ pound to the square inch of contact surface. Then, the force of adhesion $p$ between a pulley and 1 square inch of a belt 1 inch wide — varying according to the preceding — multiplied by the contact area $A$ in square inches gives the number of pounds which each inch in width of the belt is capable of transmitting. Multiplying this quantity by the velocity $L$ of the belt in feet per minute gives the number of foot pounds per minute each inch in width of the belt will transmit at that speed. Dividing the product of 33,000 times the horsepower $P$ to be transmitted by the number of foot pounds per minute each inch in width of the belt will transmit at its known speed, gives the breadth $b$ of the belt required in inches.

*Rule.* With the symbols as above, the preceding may be stated in the following relation: for the transmission of a required horsepower, the necessary *breadth of belt* is $b=\frac{33{,}000P}{pAL}$.

**High-Duty Belts Required.** Under no circumstances ought a single thickness of belting to be used on an elevator. A belt of double thickness, or even greater, should always be used. This should be of good quality leather, center stock being the best. The reason for this is that the work done by belts in the operation of elevator machinery is of the most severe nature. Even though the belts are well proportioned for the work they have to do, there are frequently occasions when they are subjected to undue strains. The sudden shifting of the belts at stopping and starting is in itself a source of

serious wear, the slippage on the pulleys being no small item. Where belt shifters are used they tend to curl up the edges of a soft or thin belt.

### Single-Belt Electric Elevators

**Standard Belt Sizes Fixed.** In the case of the single-belt electric elevator, as it is called, Fig. 44, Part I of Elevators, and Fig. 185, no calculations by the elevator maker are necessary in regard to the belt, because, in the case of motors designed for this kind of work, the width of belt and diameter of pulley are arranged at the time of designing the motor. These are standard with each size of motor.

**Size of Motor Required.** *Calculation.* The elevator builder has only to find the size of motor best adapted to do the work. This is found exactly as has been previously described, by multiplying the load in pounds to be lifted by the speed in feet per minute and dividing the result by 33,000, then adding an allowance for friction. With a worm-gear apparatus such as is generally used, the factor allowed for friction in common use among elevator men is 50 per cent. Therefore, for example, if the theoretical horsepower is 5, $2\frac{1}{2}$ horsepower must be added, and $7\frac{1}{2}$-horsepower motor would do the work, provided, of course, the weight of the cage is fully counterbalanced.

*Use of Oversizes.* However, one cannot always find a motor of exactly the horsepower calculated, as motors usually are made in sizes of 5, $7\frac{1}{2}$, 10, 15, 20, 25, etc., horsepower. Accordingly, the standard motor of the nearest higher horsepower to the horsepower calculated is the one to be used. A motor of smaller size than calculated should never be used, especially in the case of induction motors, for the torque is less at starting with an alternating-current elevator motor than with a direct-current elevator motor.

It should be borne in mind that the power expended by an electric motor is always in proportion to the work done by it, and that the use of a motor ample in size for the work required entails only a very slight additional waste of electrical energy, which is insignificant when compared with advantages gained by an absence of heating, the additional torque in starting, the surplus power ready in case of emergency, the longer life of the motor as a result of the decreased liability to heat under severe service, and the greater surface presented by the brushes and commutator segments in contact for the transmission of current.

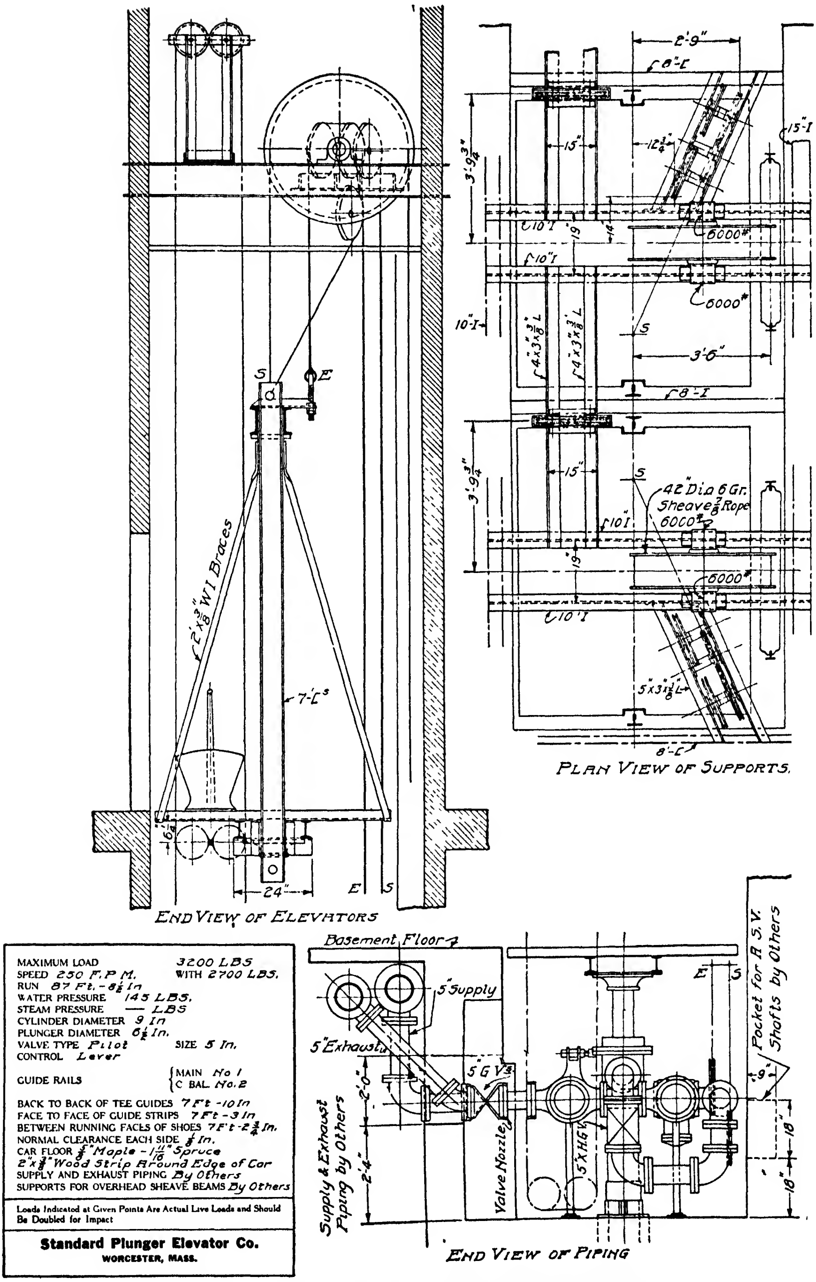


Fig. 253. Details of Hydraulic Plunger Elevator Installation
*Courtesy of Standard Plunger Elevator Co., Worcester, Massachusetts*

## HYDRAULIC-ELEVATOR POWER REQUIREMENTS

**Factors in Power Estimation.** With the hydraulic elevator the estimation of the horsepower required is very tedious. There are many phases of the subject, each one differing from the others, due to several causes — principally the water pressure to be carried, the style or type of machine used, Figs. 63 to 118, Part II of Elevators, and Fig. 253, the speed at which the car is to run, and the frequency of the trips to be made.

**Working Pressure.** It must be remembered that the immediately available supply of power is always limited, the water being stored in a pressure tank of sufficient capacity to run the elevator only a limited number of trips. The pressure is usually maintained through the medium of air compressed in the same tank above the water, and this pressure will begin to drop as soon as the elevator is started.

*Maintenance of Pressure.* The effect of a pressure drop of 2 or 3 pounds to the square inch in the pressure tank has the effect of opening the valve in the steam pipe supplying the pump, Fig. 118, Part II of Elevators. The pump then begins to supply additional water, but the elevator continuing its trip in the meantime draws off the water as fast, or perhaps faster, than it is supplied. Hence, after the car has reached the end of its travel, the pump continues to work until the water in the tank reaches the level which compresses the air to normal pressure. The pump is then automatically stopped by the closing of the throttle valve in the steam pipe. This is governed by the pressure in the tank through a poppet or cage valve.

The water used to raise the elevator car remains in the cylinder until the down trip is made, when it is discharged into another tank, from which it is pumped back into the pressure tank to be used over again. During the interval between the rise and the descent of the car, the pump has ample time to restore the pressure, provided its capacity is sufficient for the service required. It will be readily seen that where more than one elevator is supplied from the same pressure tank, the pump must be of correspondingly greater capacity. In designing a plant of this character, too great care cannot be used in selecting a pump or pumps of sufficient size to do the work. Allowance must also be made for a possible drop in steam pressure at the pumps.

**Capacity of Pumps.** After determining the proper working pressure and the quantity of water to be used for each trip, the

important requisite is to ascertain the greatest number of trips that can be run in a given time. This is determined by the time required to make 1 round trip, including stopping, loading, and unloading. When the greatest number of trips that can be made in a given time, say, 1 hour, have been ascertained, the next step is to calculate the quantity of water that will be used in making this number of trips. The selection of a pump capable of pumping the required amount of water at the pressure to be used and in the time estimated is a comparatively simple matter, as all makers of pumps supply tables giving this information.*

*Allowance for Leakage.* After having determined the theoretical quantity of water used by the elevator in a given time, and before determining the size of pump required, allowance must be made for waste of water through leakage. Pumps, like every other piece of machinery, usually operate best when first installed, and their subsequent performance is largely dependent upon the attention given them. Owing to the severe and incessant service all elevator machinery is subjected to, it is frequently a very difficult and arduous matter to keep them in prime condition. The water pistons and valves begin to leak, and their efficiency is thus impaired. Leakages are also likely to occur in the piston of the hydraulic-engine cylinder or in the operating valve. In such cases, the pump, even though itself water-tight and efficient, will not be able to keep up the tank pressure, if its capacity is figured too closely.

For these reasons it is customary, in determining the capacity of the pump, to add at least 30 per cent — some makers add 40 per cent — to the amount of water estimated to be used by the elevator or elevators. While this addition to the capacity of the pumps increases the first cost, the results always prove the wisdom of the precaution.

**Hydraulic Pumps Used.** The most serviceable and economical pumps are the compound duplex, the triple-expansion duplex, and the compound flywheel pumping engine. Of these, the compound duplex is used to a greater extent, the economical advantages of either of the others being less apparent in view of first cost, except in the larger plants.

* The student is referred to Vol. I of the Cyclopedia of Mechanical Engineering for further information on pumps.

**Horsepower of Pumps.** *Theoretical Computation.* The power required to raise water varies directly with the quantity and the height to which it is raised. To find the theoretical horsepower $P$ required to elevate water to a given height, multiply $Q$ the number of gallons to be raised per minute by 8.35, which is the weight in pounds of 1 gallon of water. This product then should be multiplied by $h$ the number of feet the water is to be raised, and the result is the power in foot pounds. Dividing this product by 33,000 gives the theoretical horsepower.

*Rule.* Stated in concise form the above relation for finding *theoretical horsepower* is $P=\frac{8.35\,Qh}{33{,}000}$.

*Actual Power.* To obtain the actual horsepower required, makers of pumps usually add 75 per cent to the theoretical power computed, although some makers double it.

**Proportioning Hydraulic Piping.** *Good Design Essential.* The piping between the pump and tanks, and the tanks and hydraulic engine is a very important feature in the economical and efficient operation of the hydraulic elevator. The pipes must be ample in area, short as they can possibly be made, and all turns and bends must be avoided as far as practicable. Where bends must be employed, they should be of large radius, for otherwise a great amount of power will be lost in friction. While this is provided for to a large extent in the estimate of the pump maker in figuring the performance of his pumps, still, badly designed piping connections between the pump, tanks, and elevator will reduce the effective service to a point which will seriously disappoint the designer of the elevator plant, both in the results attained and in the cost of operating.

*Undersizing Requires Extra Pressure.* In planning an office building where it is designed to use an elevator service operated by hydraulic pressure, the mistake is frequently made of economizing both in the space to be devoted to the pumping machinery and tanks, and also in the number of elevators assigned to the service required. When the plant is installed and found to be inadequate for the needs, the only remedy at hand is to carry a higher pressure. This expedient frequently results in an increased percentage of loss, both from the overworking of the pumps, as well as from an increased loss of total effect due to increased friction in the passage of the water through

TABLE V

**Bends for Various Pipe Diameters**

| Diameter of Pipe (in.) | Radius of Curve (in.) |
|---|---|
| 2 to 3 | 18 |
| 3 to 4 | 20 |
| 6 | 30 |
| 8 | 42 |
| 10 | 60 |

the piping. The latter loss is the more noticeable of the two, but where the piping has been designed with a liberal area, with the bends few, and the piping short as possible, an increase in pressure can be resorted to economically and with good results in an emergency where the plant has been found unequal to the demand otherwise.

**Friction in Piping.** *Factors.* Velocity is the great element in the friction of water in its passage through pipes, the pressure in no way affecting it.

From the fact that the area of a pipe increases as the square of its diameter, it naturally follows that when the diameter of a pipe is increased, say, to twice, its area is 4 times as great, but its circumference is only doubled. Hence, it is readily seen how important it is to pay close attention to having all piping ample in area or carrying capacity, in order to avoid undue friction.

*Variation.* The frictional resistance to the flow of water through iron pipes of a uniform diameter is independent of the pressure, and increases directly as the length, very nearly as the square of the velocity of the flow, and inversely as the diameter of the pipe.

**Curvature of Hydraulic Piping.** *Relative Retardation of Flow.* The time occupied in the passage of an equal quantity of water through pipes of equal lengths and diameters and under equal heads is about in the following ratio: 90 in a straight line; 100 in a true curve; and 140 in a right angle.

*Proportions.* When pipes branch off from the main, or when they are deflected at right angles, the radius of the curvature should be proportional to their diameter, as shown in Table V.

**Computation of Water-Flow Rate.** To compute the rate of flow of water through a pipe, the length and diameter of the pipe

TABLE VI

Water Flow through Pipes

| Pipe Diameter $D$ (in.) | Coefficient $K$ | Pipe Diameter $D$ (in.) | Coefficient $K$ |
|---|---|---|---|
| 1 | 4.71 | 7 | 612.32 |
| 1¼ | 8.48 | 8 | 854.99 |
| 1½ | 13.02 | 10 | 1493.5 |
| 2 | 26.69 | 12 | 2356. |
| 2½ | 46.67 | 14 | 3463.3 |
| 3 | 73.5 | 16 | 4836.9 |
| 4 | 151.2 | 18 | 6493.1 |
| 5 | 263.87 | 20 | 8449. |
| 6 | 416.54 | | |

and the head or pressure being known, the method employed is according to the rule following, using the coefficients in Table VI.

*Rule.* Divide the tabular coefficient $K$ corresponding to the diameter of the pipe by the square root of the *ratio of inclination* — that is, length of pipe in feet $L$ divided by head in feet $h$ — and the quotient will be the required volume of the flow of water $Q$ in cubic feet per minute; or, the *rate of flow* is $Q = \frac{K}{\sqrt{\frac{L}{h}}}$.

*Illustrative Example.* A pipe has a diameter of 6 inches and a length of 765 feet. What is its discharge per minute under a head of 85 feet?

The inclination equals 765 divided by 85, or 9, and the square root of this is 3. The coefficient for a 6-inch pipe in Table VI is 416.54. From these, the rate of flow equals 416.54 divided by 3, or 138.85 cubic feet per minute.

**Computation of Pipe Diameter.** To compute the diameter of a pipe when its length and the head and rate of discharge are known, use the following rule.

*Rule.* Multiply the rate of discharge $Q$ by the square root of the ratio of inclination $\frac{L}{h}$, find the coefficient $K$ in Table VI which most nearly corresponds with this product, and opposite it will be found $D$ the proper diameter of pipe. Thus, by referring the relation $K = Q\sqrt{\frac{L}{h}}$ to Table VI, the pipe diameter $D$ is found.

*Illustrative Example.* Taking the quantities of the preceding problem: 138.85 times the square root of the quotient of 765 divided by 85 — or 3 — equals 416.55. Opposite this number in Table VI is 6 inches, the diameter of the pipe.

**Pressure-Head Computation.** *Pipe Size and Discharge Known.* To compute the head when the length, diameter, and discharge are known, the method is as follows.

*Rule.* Divide the tabular coefficient $K$ for the diameter of pipe $D$, Table VI, by the discharge $Q$; square the quotient, and divide the length of pipe $L$ by it. The last quotient will be the head in feet $h$ necessary to force the given volume of water through the pipe in 1 minute; or, for a given diameter of pipe $D$ corresponding to a tabular coefficient $K$, the *pressure head* is $h = \frac{L}{(\frac{K}{Q})^2}$.

*Illustrative Example.* Applying the quantities of the two preceding problems, we have: 416.54 divided by 138.85 is 3; the square of 3 is 9; and 765 divided by 9 equals 85, the head in feet.

**Pressure of Water Column.** *Static Head.* At a temperature of 60° Fahrenheit, a column of fresh water 27.71 inches high exerts a pressure of 1 pound per square inch at its base. Hence, to find the unit pressure $p$ in pounds per square inch due to any given head $h$ of water in feet, it is only necessary to find how many times 27.71 is contained in the height expressed in inches.

*Rule.* A more simple way to find the pressure corresponding to a given head is to multiply the height in feet by .43, 1 foot or 12 inches being .43 of 27.71 inches; thus, the *unit pressure* in pounds per square inch is $p = .43\ h$.

*Illustrative Example.* Taking, for example, the head of water assumed in the last examples — 85 feet — this head multiplied by .43 equals 36.55 pounds pressure per square inch. This applies to a column 85 feet high when at rest.

*Variable Head.* Should water be drawn from the standpipe even though it is kept filled from a tank above, the pressure will drop, as soon as the water commences to move, unless the pipe is adequate to keep up the supply without friction. To do this the pipe must be ample in area.

The same conditions surround the flow of water through pipes from a pressure tank where the pressure is maintained by a reservoir

of air, the latter being under pressure and filling the space above the water, except that as soon as water is drawn from the tank, the air begins to expand as more room is afforded it. The pressure falls off slightly as the air expands, until the automatic appliance before described starts the pump and so replenishes the deficiency; but the pump will not supply any drop in pressure which is due to friction in the pipes.

**Calculations of Weight and Volume of Water.** *Standard Units.* The United States standard gallon contains 231 cubic inches, and 1 gallon of fresh water at 60° F. weighs 8.33 pounds. There are 7.48 gallons in 1 cubic foot of water at 60° F. and 1 cubic foot of water weighs 62.32 pounds at that temperature.

*Circular Containers.* To find $Q$ the contents of any vessel, such as a cylinder, for example — either that of a pump or of a hydraulic engine — it is only necessary to multiply the internal cross-sectional area $A$ in square inches by the length $l$ in inches and to divide the product by 231 for United States standard gallons, or by 1728 for cubic feet. Thus, where the cross-section $A$ and length $l$ are in inches, the *volumetric contents* of such a vessel, in gallons, are $Q=\frac{Al}{231}$; and, in cubic feet, $Q=\frac{Al}{1728}$.

To find the area $A$ of a circle in square inches multiply the square of the diameter $D$ by .7854; or, where the diameter $D$ is in inches, the *circular area* in square inches is $A=.7854D^2$.

*Weight of Water in Pipe.* To find the weight in pounds $W$ of water in a pipe, multiply the product of the length $L$ of the pipe in feet and the square of the diameter $D$ in inches by .34; or, the *weight* of water is $W=.34D^2L$.

*Weight of Water in Tank.* To find the weight $W$ in pounds of water in a rectangular tank, multiply the product of the length $l$ in inches, the breadth $b$ in inches, and the height $h$ in feet by .43; or, $W=.43\ blh$.

If the tank is circular, multiply the product of its circular cross-section $A$ in square inches and its height $h$ in feet by .43; thus, the *weight* of water is $W=.43\ Ah$.

**Head of Water Corresponding to Pressure.** To find the head of water in feet $h$ corresponding to a given unit pressure $p$ in pounds

per square inch, multiply the pressure by 2.309. This comes from the ratio which the height — 27.71 inches — of a column of water exerting a base pressure of 1 pound has to a head of 12 inches or 1 foot — thus, $\frac{27.71}{12}=2.309$. So, with reference to the pressure per square inch at the base the corresponding *head* of a column of water in feet is $h=2.309p$.

**Water Consumption.** *Leakage through Cylinder.* The hydraulic engine is of itself a reliable water meter, for if there is any leakage in the piston or the stuffing box — it will immediately become apparent at the cage or car, which will then creep at the landings when stopped. If the leakage is at the inlet cup of the valve, the car will creep up; if at the discharge packing, or at the piston or stuffing box, the cage will move slowly down after having been stopped. In order to have efficient service, the engine must be fairly tight.

*Displacement.* To find the amount of water used per round trip, it is only necessary to compute the amount of water used on each ascending trip, for it must be remembered that water is supplied the piston for only one direction of running. In estimating the amount of water, the bore and length of cylinder are not taken into account. The area and stroke of the piston or plunger are the factors which determine or measure the amount of water used, the cylinder, of course, being somewhat larger. In the case of the plunger machine there is a water space all around the plunger, and a certain portion of it, of course, never leaves the cylinder. When a piston is used, although it fits the accurately bored cylinder closely, there is a space at each end between the piston and the heads, and the piston itself occupies some space. From this it will be readily seen why it is necessary to measure the diameter and the stroke of the plunger or piston. This gives accurately the amount of water received and displaced each full round trip.

The actual amount of water required to supply one or more hydraulic elevators and the horsepower necessary to pump the water at a given pressure can thus be readily ascertained. The determining of the proper area of the plunger or piston is a different matter, and is one which has to be considered carefully in designing the machine.

**Determination of Piston or Plunger Area.** *Buoyant Effect of Plunger.* Those plunger elevators which have the platform attached to the top of the ram or plunger have no multiple, but the plunger travels the entire height of the run, Fig. 253. Where the travel is great, the lifting capacity of the elevator varies to a considerable degree according to the extent to which the plunger is immersed because of the buoyant effect of the water. This variation in lifting capacity, which is peculiar to the plunger type of hydraulic machine, is a defect which admits of no remedy, although it can be, and is, counteracted to some extent. It might be entirely overcome by the use of chains connecting the car with the counterpoise weight instead of the wire cables which are now used, the chains being made sufficiently heavy to counteract the variations in effective weight of the plunger. However, the noise made by chains bars their use and consequently, when estimating the area of a plunger machine of this type, it is necessary to take the weight of the plunger and the buoyant action of the water into consideration.

*Counterpoising.* The weight of the plunger could be counterpoised, but not entirely, because if the plunger were fully counterbalanced it would not be able to descend when the car nears its lower level, through inability to displace the water remaining in the cylinder. For example, suppose a plunger weighing 2700 pounds displaces an amount of water weighing 1350 pounds. It is clear that the plunger could not be counterbalanced even to within $\frac{1}{2}$ its weight. Hence, between the upper and lower landings there would be a difference of this amount in its lifting capacity. This is not an extreme case, but simply a case of a high run. It is given here to show one of the disadvantages accompanying a long travel with this type of elevator. Of course, with a shorter or lower travel the discrepancy would not be so great, but whether it is much or little, it must be estimated when designing a machine of this kind.

*Use of Multiple.* Where a multiple is used and the travel of the plunger or piston is only a fraction of that of the car, the conditions described above do not exist. In the case of the vertical machine, Fig. 97, Part II of Elevators, the plunger or piston, as the case may be, itself acts as a counterpoise, and is utilized as such, being estimated in calculating the proper diameter of piston.

With the horizontal machine, the weight of the piston and sheaves is not available as a counterpoise, and they really figure as part of the friction, being here a dead weight which has to be moved in either direction and which does not yield any compensating advantages.

The multiple of these machines varies with the type and general conditions surrounding their installation and service. The vertical machines are geared from 2 to 1 up to as high as 8 to 1, while the horizontal machines usually are geared either 8 to 1 or 10 to 1.

*Rule for Calculation of Piston Area.* The method of ascertaining the diameter of piston, the pressure being known, is as follows: first, add to the load to be lifted the weight of the unbalanced portion of the cage; then multiply this result by the ratio of the gearing; and to this product add a certain per cent for the power required to overhaul the cables around the sheaves and, in the case of the horizontal machines, to move the piston and sheaves horizontally on the ways on which they travel. This additional amount differs according to the size and weight of the sheaves, crosshead, and piston, and according to the number, diameter, and length of the cables used; there is no rule for determining this quantity, but it may be stated that it varies from 15 per cent to as much as 40 per cent of the power required to lift the load. Most builders of horizontal hydraulic machines commonly used for their guidance tables calculated from data collected during their practice in this line, but, as the horizontal machine has practically gone out of existence, or at least is not being made now, the matter is not of vital importance. The ratio of gear reduction is an important factor both in the area of the piston and of the speed.

Accordingly, it is usual to take the total load $W$ to be lifted, including the weight of the cage, and multiply it by the ratio of gearing $r$; add to the product from 15 to 40 per cent of itself; and divide the result by $p$ the pressure of water to be used. This gives the area of piston $A$ in inches, from which the diameter $D$ is obtained by the use of tables, or by dividing $A$ by .7854 and taking the square root of the quotient. The equation for this process which may be used in finding the hydraulic-piston diameter is $D=\sqrt{\frac{A}{.7854}}$, in which the *piston area* is $A=\frac{Wr+(15 \text{ to } 40\%)Wr}{p}$.

## ELECTRIC-ELEVATOR POWER REQUIREMENTS

**Selection of Motor.** In determining the horsepower required for the operation of a direct connected electric winding engine, Figs. 126 to 172, Part III of Elevators, and Fig. 254, the method previously described for the belt-type electric elevator is preliminary. Having settled by this method the size of motor required, the next step is to select a motor having a suitable speed.

Usually the elevator builder has his winding machine so designed that he can use gears and worms of two or three different pitches in the same gear casing, the number of teeth in the gears differing enough to give an opportunity for a proper selection. This is further assisted by using a drum of suitable diameter, the greatest difficulty in making a suitable motor selection being with the slow-speed elevators, such as those running at from 50 to 100 feet per minute.

**Motor Speed.** To illustrate; let it be assumed that it is desired to lift a load of 4000 pounds at a speed of 75 feet per minute. It is customary to over-counterpoise these machines anywhere from 500 pounds up to 40 per cent of the maximum load to be lifted.

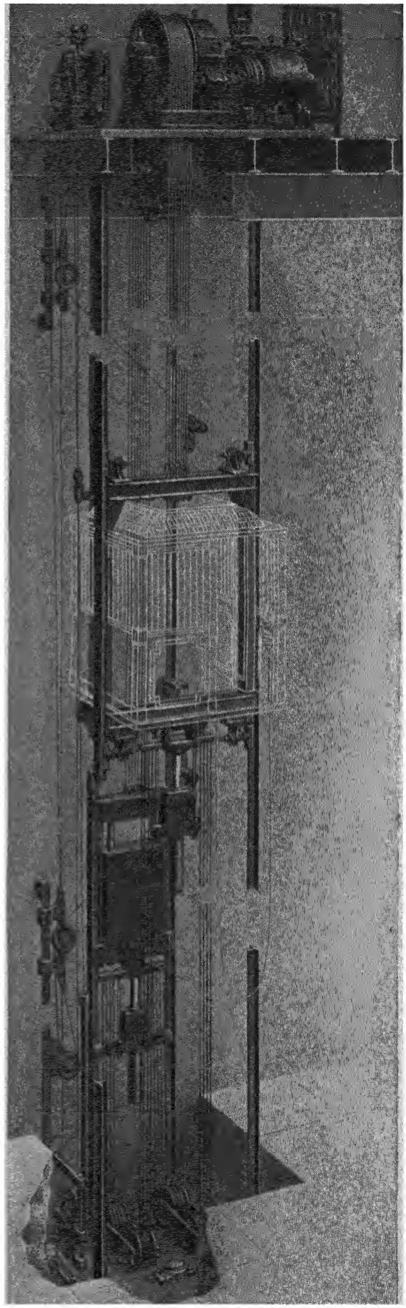

Fig. 254. Gurney Electric Traction Installation

*Problems of Horsepower. 1.* Some makers estimate the horsepower on the basis of the overload, deducting this from the maximum load to be lifted. Thus, 40 per cent of 4000 pounds is 1600 pounds, which, deducted from 4000 pounds, leaves 2400 pounds as the effective load to be lifted. This result multiplied by 75 feet per minute is 180,000 foot pounds per minute. Dividing this by 33,000, the theoretical horsepower required is found to be 5.46. This is then doubled, giving about 11 horsepower as the size of the motor required. But the nearest standard motor to this size is rated at 10 horsepower. Some makers would use a 10-horsepower motor in such a case, and the motor would be found to do the work, although at a decided disadvantage. It would, therefore, be better to use a 15-horsepower motor and to have a certain amount of reserve power always at hand.

*2.* Another and better way of arriving at the required power is to take the full load to be raised, making no deductions for extra counterpoise, and add 50 per cent for friction, etc. Taking the quantities of the previous problem, it is found that 4000 multiplied by 75 is 300,000, and this divided by 33,000 gives 9 horsepower. Adding 50 per cent, $13\frac{1}{2}$ horsepower is obtained. This result manifestly would require a 15-horsepower motor, and whatever over-counterpoise is used will be an advantage. There is also the advantage of knowing that the motor will do the work easily if the counterpoise only offsets the weight of the car or cage. Any excess counterpoise will operate to reduce the current consumption.

*Practical Speed.* There is no standard speed for each horsepower, the speeds differing with each maker. Each builds motors with two or three speeds in each of the standard sizes. These are designated as high speed, medium, and slow speed. In the case of a 15-horsepower motor, the purchaser would find them listed as running at 1200, 800, and probably 450 revolutions per minute.

In the present case, the motor running at 1200 revolutions per minute is out of the question on account of the abnormally large gear or small drum necessary to reduce to the speed of car required, and because of the lack of torque in starting. Hence, the choice lies between the motors with speeds of 800 and of 450 revolutions per minute.

**Drum Size.** A winding machine suitable for a load of 2 tons would probably have a worm gear with 80 teeth; and 800 revolutions

per minute divided by 80 gives 10, the speed of the drum in revolutions per minute. Dividing 75 feet per minute, which is the speed of the load, by 10, the quotient is 7.5, which is the circumference of the required drum in feet; this drum would be about $28\frac{1}{2}$ inches in diameter. As the cables used for a load of two tons would be $\frac{5}{8}$-inch in diameter, this size of drum would be suitable, in view of established rule among users and dealers in cables that the drum or sheaves over which they run should be at least 40 times the diameter of the cable.

**Counterpoising Effect on Gears and Cables.** In selecting gears and cables for a direct-connected electric elevator, a finer pitch of gear and smaller diameter of cable may be used than those which would be required for a 2-belt power driven elevator, because of the extra counterpoise which frequently is 40 per cent of the load. This tends to lighten the burden on both gear and hoist cables; and the counterpoise directly from the car, Fig. 150, in Part III of Elevators, and Figs. 186 and 207, is a prime factor in that it relieves the hoist cables of everything except the actual load on the platform plus 500 or 600 pounds. In order to insure the car's descent by gravity, the counterweight attached directly to the crossbeam of the car is always less in weight than the actual weight of the cage by about 500 or 600 pounds. This allowance is not much in a building of 4 or 5 stories, but in a building of 10 or 12 stories it would not be enough, owing to the weight of the cables connecting the counterpoise weight to the car.

*Counterbalancing by Hoist Cables.* Hoisting cables $\frac{5}{8}$-inch in diameter weigh approximately $\frac{3}{4}$ pound per foot of length, and if the building is 10 stories, or 130 feet in height, two $\frac{5}{8}$-inch cables of this length would weigh nearly 200 pounds. This makes a considerable addition to the amount of weight balancing that of the car, and does not leave a sufficient margin of unbalanced weight of cage to enable it to descend promptly when at the top of its travel. Of course, this disadvantage exists only while the cage is at the upper stories, for, as the car descends and the cable gradually travels over the sheaves from the weight slide into the hatchway, the latter portion soon balances the former, until, when the car is at the lowest point of its travel, the conditions are reversed and the preponderance of weight is with the car. But when this difference in the unbalanced

weight of the cage does not exceed the amount just described, the simple remedy is to leave as much as 800 pounds of unbalanced weight in favor of the cage. The immediate effect is only on the hoisting cables — in imposing a slightly greater burden on them — while the ultimate effect on the operation of the machine is decreased by the counterpoise attached to the winding drum. This counterpoise is in addition, being made heavy enough to take care of the unbalanced portion of the weight of the cage.

*Chain Counterbalance for Operating Economy.* When close attention to the economical consumption of current is an object, the use of chains in the hatchway in connection with the cage is resorted to, the method of this application having been fully described.

*Stress and Wear.* It will be clear to the reader, from what has been said about counterpoising in this connection, that there is a great difference in the amount of stress and wear on the gear and worm of the electrically operated elevator and of the two-belt elevator driven from a line shaft. In the latter, the amount of power consumed is not so carefully considered; the initial cost for purchase and installation is lower and less attention is paid to the matter of counterbalancing, and so there is a greater duty imposed on the worm and gear. Where the cage is fully counterpoised, or partly counterpoised as in the case of a direct connected machine, the gear has less to do in lifting a full load, and when the loads lifted average about 50 per cent of the full capacity of the elevator the work of the gear and worm is very light indeed. Moreover, when lowering empty or with light loads, the wear is on the opposite sides of the teeth of the gear and the threads of the worm, so that gearing of a finer pitch may be used safely. These favorable conditions do not exist in connection with the two-belt power elevator, in which all of the load lifted is always on the gearing.

**Motor Calculation for Traction Elevator.** In the traction type of electric elevator, Figs. 151, 165, and 168, Part III of Elevators, and Fig. 254, the simplification or absence of gearing eliminates a very large percentage of the friction which is inseparable from the worm-and-gear form of machine, but the wrapping of the cables around the drums and the loads at either end of the cables produce a certain amount of pressure on the journals and boxes or bearings which cannot be ignored. This has been variously estimated at

from 10 to 15 per cent of the effective power required to operate the machine. It is customary in estimating the power capacity first to estimate on the basis of using a counterpoise equal to the weight of the cage and cab combined plus 25 per cent of the maximum load to be lifted, and then to figure the theoretical horsepower required to lift this load at the speed desired, and finally to add 25 per cent to the result.

*Illustrative Example.* For example, suppose it is desired to lift 3000 pounds at a speed of 400 feet per minute. Then, the effective load lifted is 75 per cent of 3000, or 2250 pounds, which multiplied by 400 gives 900,000 foot pounds per minute, and 900,000 divided by 33,000 gives 27.3 horsepower. Adding to this 6.8 — or 25 per cent of 27.3 — makes 34.1 horsepower. Therefore, a 35-horsepower motor would do the work.

**Provide Surplus Power.** In practice it has been found best at all times to estimate for a surplus of power rather than for just sufficient to do the work, and a liberal estimate is frequently cause for congratulation when some unforeseen difficulty appears later on.

**Electrical Quantities.** *Pressures and Currents in Use.* The voltage of electric current generally in use for the operation of elevators is 110, 220, or 500 volts. The amount of current equivalent to 1 horsepower is 6.78 amperes for 110 volts, 3.39 amperes for 220 volts, and 1.49 amperes for 500 volts pressure. In making the preliminary or approximate estimate of current required, it is customary to figure 8 amperes for 110 volts, 4 for 220, and 1.75 for 500 volts.

*Equivalents.* The following equivalents are useful in power determinations:

Volts $= E =$ pressure $= C \times R$ Watts $= W =$ voltamperes $= C \times E$

Amperes $= C =$ current $= \frac{E}{R}$ Ohms $= R =$ resistance $= \frac{E}{C}$

1000 watts = 1 kilowatt 746 watts = 1 mechanical horsepower

Horsepower × 746 watts ÷ volts = amperes

Electrical horsepower = amperes × volts ÷ 746 watts

**Use of Alternating-Current Motors.** *Practical Conditions.* It may be well at this point to explain that all the foregoing examples in finding the horsepower required to operate an electric elevator are

based on the assumption that direct current is to be used as the operating force. At the time of the introduction of the electric elevator that current was the more available. Later on, when for economic reasons the alternating current became more frequently met with as a source of both power and lighting, elevator men began to turn their attention to the designing of an elevator motor which could be run by alternating current.

The principal difficulties were two: (1) reversibility; and (2) torque in starting. These are two very essential features in a good elevator motor. The latter difficulty has not as yet been fully overcome, and the first one only with the two- and three-phase current. Although innumerable experiments have been made extending over a number of years, it is safe to say that up to the present time no successfully reversible motor has been designed to run on a single-phase current; hence this type of motor is not considered.

*Two- and Three-Phase Motors.* Fair success has been attained with two- and three-phase current motors, in as much as they can be made to reverse without difficulty and to run with a fair degree of economy. Their speed, however, is variable under load. Of course, their synchronal speed differs with the number of poles in the stator —the speeds most favored by the elevator man being 900 and 720 revolutions per minute. These speeds usually drop about 5 to 7 per cent under load, and in estimating sizes of gears and drums to use with these motors this must be borne in mind. Their lack of torque in starting can be overcome by using a motor of 30 per cent greater horsepower than would be required for direct current.

*Direct-Current Motors Required for High-Speed Elevators.* The fact that the speed of the A. C. motor cannot be governed like that of the D. C. motor unfits it for use with high-speed elevators. With the high-speed elevator running at from 350 to 550 feet per minute, and by the use of a D. C. motor with interpoles, the fields can be strengthened so as to reduce the speed to almost anything desired. The increased torque they possess at the lower speeds has been made use of in lifting heavier loads than at the higher speed, and in the case of the traction elevator, in order to get the desired traction between the cables and hoisting drum, a heavier weight is used for the time, being attached to the bottom end of the regular weight, and when not in use lies in the bottom of the pit at the foot of the run.

## SAFETY DEVICES

**Safeguards Necessary.** The operation of the elevator, like that of any other machine, is never at any time entirely devoid of liability to mishap, either to the machine itself or to those using it. Ignorance and carelessness are doubtless the principal contributories. Numerous safeguards have been devised and applied for the mitigation if not complete prevention of these dangers, and it is the purpose of this section to enumerate and describe them. To better enable the reader to obtain a comprehensive idea of the number of these appliances, there is given herewith a list of them, before proceeding with the description.

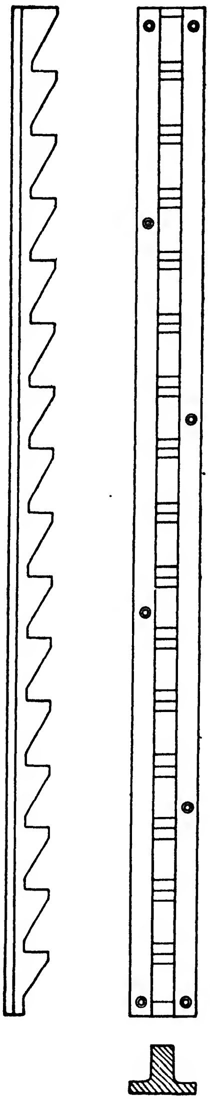

Fig. 255 Iron-Rack Guide

*Devices Employed.* Safety dogs; safety grips; numerous cables instead of a single cable, as at first used; automatic stop; slack-cable stop; centering line; air cushions; screens under overhead machinery; screens on top of cars; guards or inclosures around cars; guards or inclosures around hatchways; Meeker doors; self-closing doors; collapsible gate; locking of freight cars when loading heavily; door locks; door switches; electric brake; oil and spring buffers; bumpers; drum brake; oiling devices; and careful and thorough inspection.

### CAR SAFETIES

**Early Iron-Rack Guide.** Doubtless the first danger that presented itself to the maker or user of the elevator was that of the breaking of the lifting cable, and very early in the use of the machine efforts to prevent the falling of the cage were made. The first comprised the use of iron ratchets for the guides. These were made of cast iron and were attached to the guide posts proper. Fig. 255 shows this ratchet, or rack as it was called by the makers, also its application to the guide post. Fig. 256 shows the upper part of the car with the wrought-iron dogs which,

actuated by a spring, were intended to catch in the teeth of the racks in the event of the breaking of the hoisting cable.

The elevators in use at the time of the use of this form of safety were slow running affairs seldom exceeding 60 feet per minute and were lifted by but one cable. Of course, much depended upon the stiffness and temper of the spring which was to actuate the dogs,

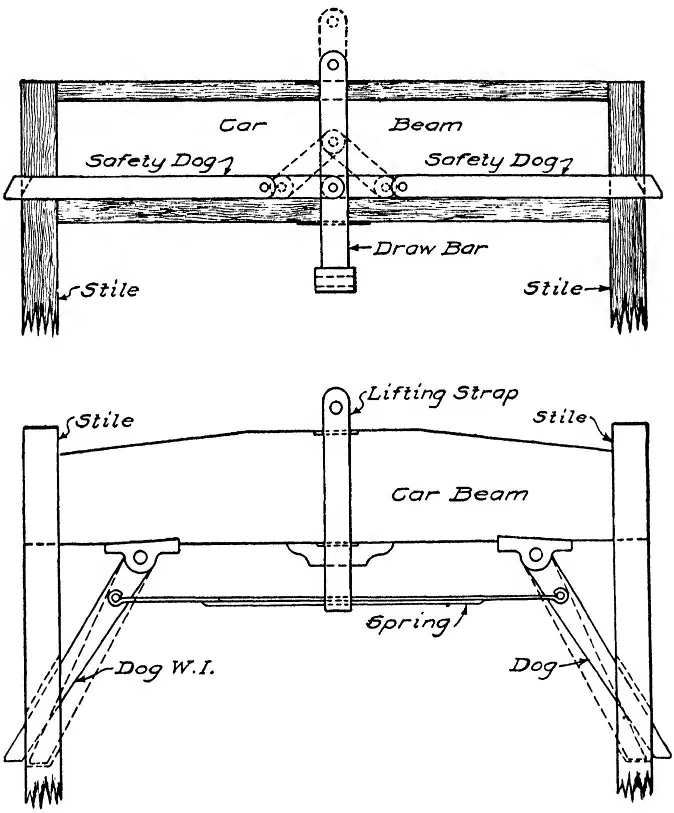


Fig. 256. Arrangement of Safety-Rack Dogs on Car

and probably no one outside the business of making elevators at that time realizes the amount of care that was bestowed on the making of these springs nor of the number of cases in which they proved their efficiency. It may be a source of surprise to the reader to find that this form of safety was almost the only one in use until a comparatively late date, 1879, when other safeties had been devised — viz, the

automatic stop, the slack cable stop in 1875, and the use of more than one lifting cable a little later.

**Eccentric-Cam Grip.** The rack guide was in general use up to the date just mentioned, when Mr. Moore of the firm of Moore & Wymans, of Boston, devised an eccentric cam to grip the maple guide. This was operated through connection with the governor as shown in Fig. 257. This type had almost entirely superseded the use of racks on the hand elevator but had not been used with

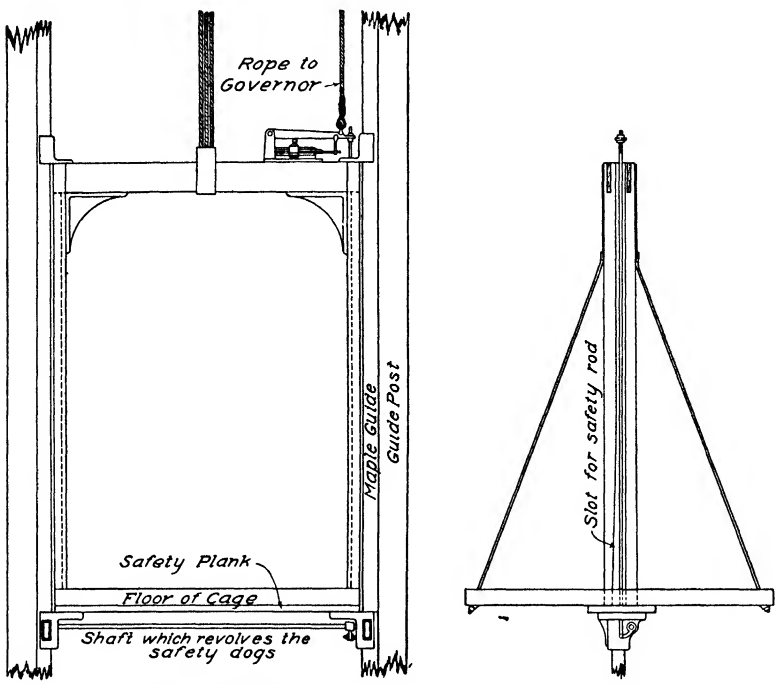


Fig. 257. Arrangement of Safety Cams and Connections on Car

cars operated by power up to this time, but once introduced they quickly displaced iron racks with all types of machines. Fig. 258 shows the form of grip or dog first used with the maple guide.

*Chisel Dog Improvement.* In operation the many small notches in the first form of cam filled up and reduced its efficiency, so the simple and effective chisel dog, Fig. 259, was devised to replace it.

*Cam for Steel Guides.* For use with steel guides, however, the form shown in Fig. 260 is satisfactory.

*Position below Platform.* It will be noted that the application

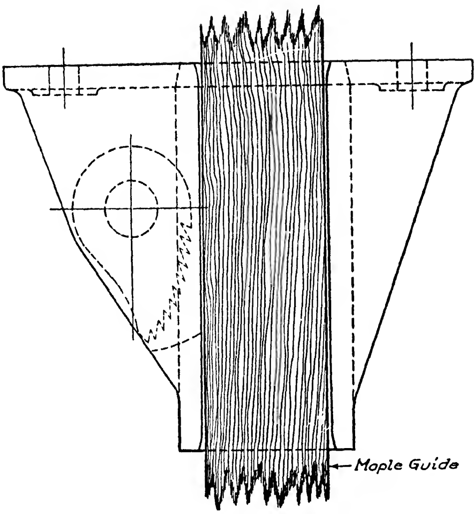


Fig. 258. Eccentric-Cam Grip for Wood Guide

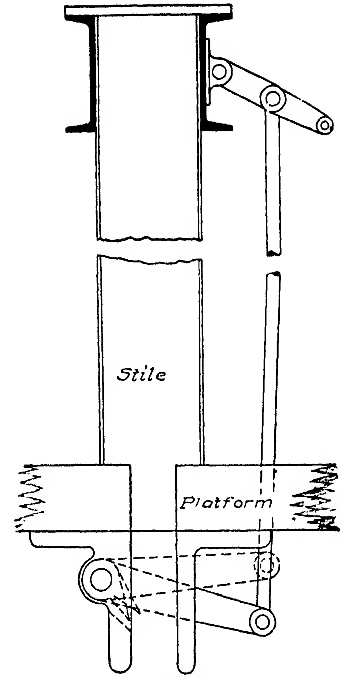


Fig. 259. Safety Chisel Dog

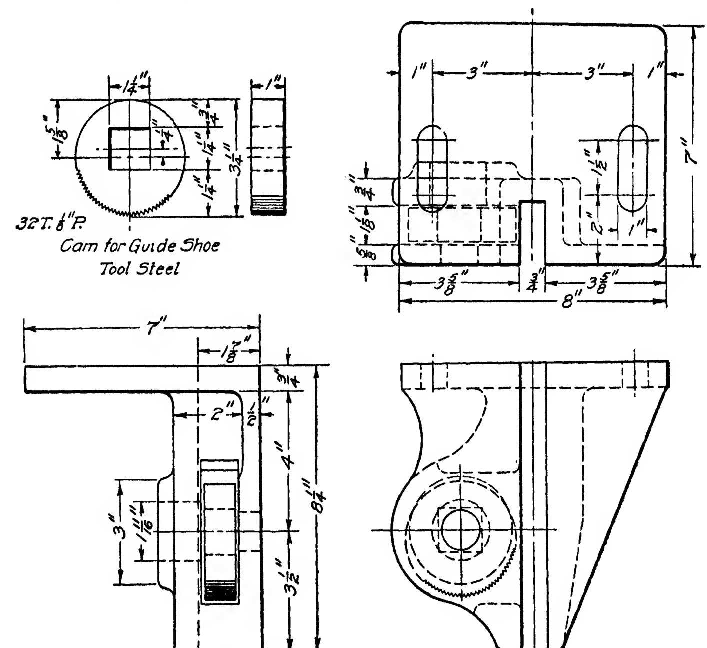


Fig. 260. Eccentric-Cam Grip for Steel Guide

of these dogs was made below the platform instead of at the upper part of the car. This was considered an improvement in itself because it was believed that the sudden stopping of a heavily loaded car would have a tendency to rupture the stiles and allow the loaded platform to drop. So it was thought that the application of the dogs below the entire load was more safe — everything depending entirely on the dogs, and nothing intervening.

### Friction Devices for High Speeds

**Development of Friction Safety.** The high speeds attained by the hydraulic elevator at the time when this machine was most popular showed the necessity for a grip which, in the event of the car falling, would stop it more gradually. The friction safety was the result. Originally it was designed for and used with the maple guide, and it was substantially the same as it is made today except that the jaws which gripped the maple guide were longer. They thus presented a greater surface to the guide, so as to prevent their sinking into the maple guide and stopping the car too suddenly.

**Modern Friction Safety.** *Grip Operation.* The modern friction safety is shown in Figs. 254, 261, 265, and 271. There are various types of this form of safety, and they are essentially the same in both principle and structure. A right- and left-hand screw is used to operate either cams or toggle joints in opening the long ends of tremendously strong tweezers, the short ends of which grip the guide sufficiently tight to hold the load. To do this it becomes necessary to make the right- and left-hand screws revolve a number of times, and this operation occupies some little time relatively, so that the car is brought gradually to a stand while falling a distance of from 5 to 10 feet, according to the adjustment of the safety.

*Comparative Efficiency.* It has been found that a car traveling 600 feet per minute may be stopped within the distance just mentioned without serious inconvenience to the passenger, while with the old type of instantaneous stop — that of the dogs actuated by a spring — the effect, if the car was running down more quickly than 100 feet per minute, was exceedingly unpleasant, if not hurtful. Of course a friction safety could not be operated by a spring.

**Governor Actuation of Friction Safety.** *Old Governor Principle.* In that portion of this treatise which describes the power elevator

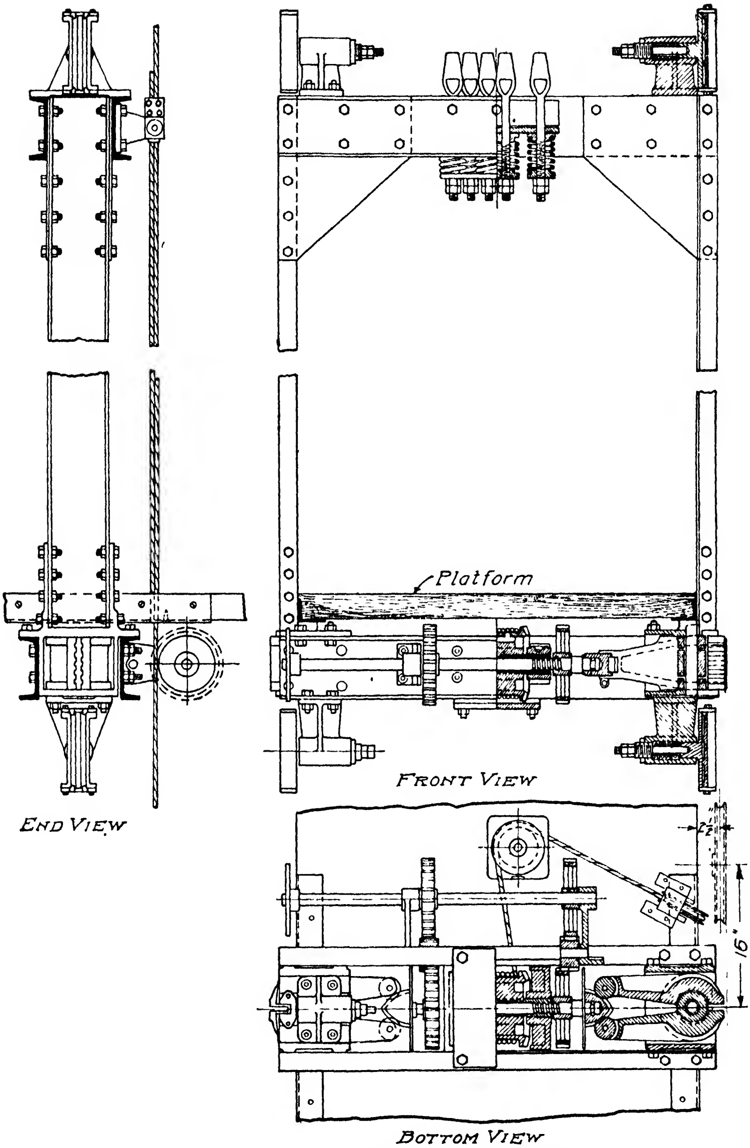


Fig. 261. Details of Drum-Type Friction Safety Installation on Car
*Courtesy of The Warner Elevator Manufacturing Company, Cincinnati*

mention is made of a governor which was used by the Otis Brothers back in the early 60's for tripping a trigger which released a weighted

lever, which in turn applied a powerful brake to a drum overhead. This principle had been resurrected, and the same or a similar type of governor had been applied to the beam of the car. This governor was actuated by a rope fixed at each end at top and bottom of the

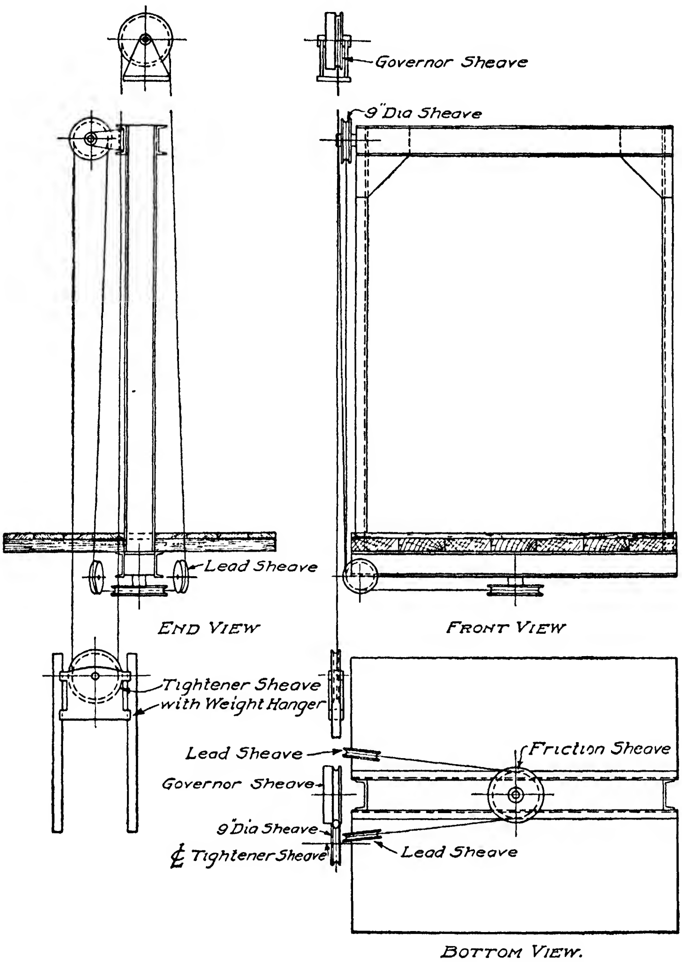


**Fig. 262. Diagram of Friction Safety Connection to Governor**

hatchway and serving as a belt to drive the governor when the car was in motion, and the governor was so arranged that when the speed of the car exceeded a certain limit the governor would trip a spring, which in turn would throw the dogs. This was an improve-

ment over the old way of waiting for the cable to break to release the spring.

*Present Automatic Governor Arrangement.* The introduction of the friction safety made necessary another form of governor for its operation, and the governor was transferred from the beam of the car to a permanent position in the attic or penthouse, as shown in the installations in Figs. 150 and 151, Part III of Elevators, and in Fig. 254, and the diagram of connections, Fig. 262. The rope which acted as a belt to drive it was made longer, and it was passed around a sheave at the bottom of the run and up over the governor sheave,

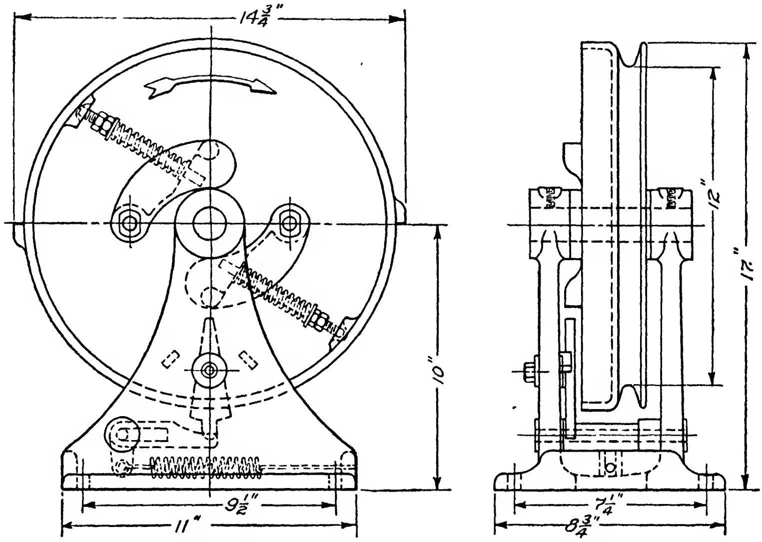


Fig. 263. Details of Friction-Safety Governor

both ends of the rope being joined. A clutch near the beam of the car held the rope sufficiently tight to cause the latter to travel with the car, but not so tight that a strong jerk would not release it from the clutch which held it. This arrangement caused the rope to travel with the movement of the car, and it in turn caused the governor to revolve at the same speed the car traveled. The governor, Fig. 263, being placed at the top of the run, was fitted with a stop which gripped and held the governor wheel whenever a certain speed was exceeded. When this occurred, the V-shaped groove in the governor sheave would hold the rope tight and release it

from the clutch on the car, the further movement of the car causing the governor rope to revolve the right and left screws, tightening the grips on the guide rails until the car was brought to a standstill. The sheave at the bottom of the run, Fig. 264, was made to slide vertically in short ways, and the frame in which the sheave was mounted was weighted enough to keep the governor rope in a state of tension sufficient to take up any stretch and also to cause it to drive the governor when the car was in motion. The rope used was of manila.

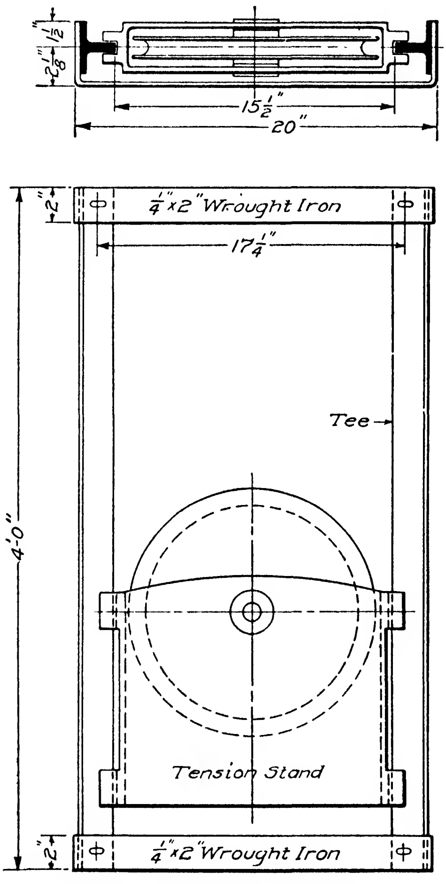


Fig 264 Tension Sheave for Governor

In some cases, Fig. 261, a shorter rope attached to and passing a number of times around a spool was spliced into the main driving rope of the governor, and when this latter was stopped and the movement of the car continued, the effect was to unwind all the rope on the spool, causing it to revolve. In this case, the right and left threads were cut in the hubs of the drum, and the screws themselves remained stationary; in other cases, the ends of the screws were keyed into the hubs of the drum, and they were made to revolve; and in still other cases, Fig. 265, the ends of the screws were fitted with miter gears, a third miter being set in between them. To this latter was attached a sheave with a V-groove running horizontally, and, instead of the short piece of rope before mentioned, the governor rope itself was led over and under leading sheaves beneath the bottom of the car and

around this horizontal sheave which, through the medium of the miter gears, caused the screws to revolve and thus apply the grips to the rails.

These various methods were all means to the same end, differing only with the ideas of the makers; and this arrangement is what is in use today, the only change being in the substitution of a wire

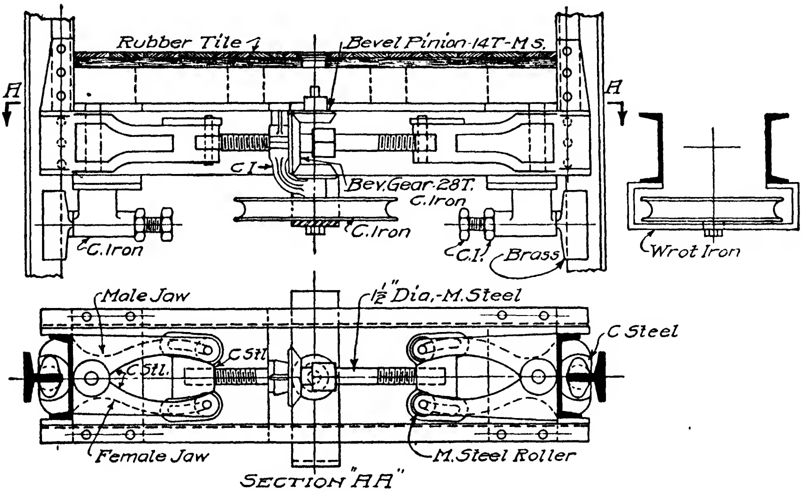


Fig. 265. Details of Miter-and-Sheave Type of Friction Safety

rope for the manila where the runs are long and the consequent stretch of a manila rope considerable.

## CABLE SAFETIES

**Multiple Cables.** When the hydraulic elevator was at the height of public favor, and while the safety just described was gradually developing, it occurred to Mr. W. E. Hale, then a prominent manufacturer in this line, that a multiplicity of lifting ropes would be a better and more reliable safeguard against a falling car than that of placing any dependence on an appliance which might become deranged and fail to work in an emergency. The idea in connection with using a number of lifting ropes being that hardly any combination of circumstances would occur which would cause them all to break at once, each rope when new being capable in itself of sustaining the whole load. Consequently, as many as six cables were used on each passenger elevator, and two or three on the freights.

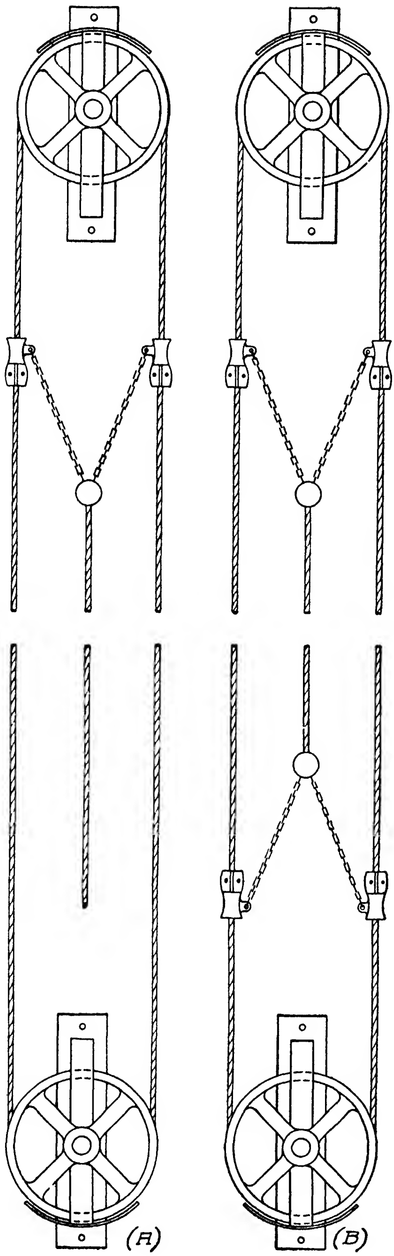


Fig. 266. Centering Lines

*Suitability for Traction.* This arrangement was easily attained in the hydraulic elevator because the cables ran on sheaves altogether. With an elevator in which the lifting cables were wound and unwound on and off a drum, the width of face required for six lifting cables made it impracticable, so in such cases only two were and are now used. In traction elevators, Fig. 151, Part III of Elevators, and Fig. 271, the six cables are used both on account of this feature of safety and also to give better traction, and the practice has extended to multiple cables for the weights, as well as for the cars.

**Automatic Stop.** Another need which made itself manifest early in the use of the elevator was that of a means of stopping the car automatically and independently of the operator at the end of its run, either way.

*Cable Type.* The first safety of this nature was the striker attached to the car which traveled with the motion of the car up and down the operating cable, striking a button set on it at either end of the travel and so bringing the machine to rest, as has already been described in connection with Fig. 81, Part II of Elevators, under the head of Auxiliary Devices.

A modern development in which a cable set diagonally near the head and foot of the shaft is operated by a deflecting sheave on the car is shown in Fig. 254.

*Self-Contained Types.* The liability of breakage of the operating cable with the previous arrangement developed the necessity of having a stop which was part and parcel of the engine itself, and led to a variety of expedients, each particular type of machine being supplied with its own particular form of automatic stop, such as hydraulic limit valves and electric hatch switches, all of which have been described in their appropriate places — Figs. 83 to 89, Part II, and Figs. 145 and 151, Part III of Elevators, and Fig. 271.

**Centering Line.** Early in the days of the two-belted power elevator, an appliance that would surely and effectually shift both belts on to their respective loose pulleys was devised in the form of the centering line, Fig. 266, which has also been described. The usefulness of this contrivance has not ceased, for it is extensively used today on certain types of electric freight elevators to center the operating switch.

**Slack-Cable Stop.** *Necessity in Obstructed Hatchways.* Today, in almost every state in the Union, there are laws governing the installation and operation of elevators. Among them are those which compel the inclosing of all elevator shafts with a fireproof inclosure which serves the double purpose of a guard against accidentally falling down the hatchway, as well as that of a protection against the spread of fire. Formerly these inclosures were only installed as a matter of comfort or convenience. A line of hatchways in a building creates a draft of air which, in cold weather and with doors open in the lower stories, is the cause of much discomfort. To prevent this was the only cause of the use of inclosures which then were made of wood for economy.

When these inclosures were not used, a guard of wooden bars was placed around the hatchway on each floor for the freight elevators, and for the passenger machines there was used a wire screen about 7 feet high, with an apron of the same material extending to the ceiling opposite the door in the cab. Under such conditions, it was not uncommon for goods to be piled around the hatchway in such a manner as to obstruct the passage of the car. If it came in contact with them in ascending, it simply pushed them aside, but if descend-

ing, it would become caught and stopped in its descent, while the engine or winding gear would continue in motion and would unwind all the cable. In such case, it would be necessary to rewind the cable on the drum before removing the obstruction which held up the car, but unfortunately this precaution was not always thought of, and, when the obstacle was removed, the car would drop to the bottom of the run, doing considerable damage.

The safety dogs at that time were mostly operated by springs and were not intended to act unless the lifting cables became broken. The tension on them in being drawn swiftly over the upper sheaves prevented the springs from throwing the dogs which were designed to arrest the fall of the car. It was a consideration of these conditions which prompted the construction and application of the slack-cable stop.

*Usage.* The writer was the first — 1874 — to devise and apply this form of safety. It was not patented at the time, and its usefulness was unrecognized by the makers of elevators for two or three years, but today it is in general use, specifications for the installation of elevators invariably calling for its employment. It is used now in various forms, each different type of elevator having its own device, all of which have been fully described in preceding chapters.

## INCLOSURES

**Shaft Inclosures.** Inclosures around hatchways usually are very expensive affairs, frequently costing as much as the elevator itself, and, while they are not really a part of the elevator proper, they are so closely related to it that a brief description of them will not be out of place.

The word *hatch* is a nautical term and means an opening in the floor or deck through which goods are handled. The term doubtless was borrowed at the time when the elevator was first instituted, as were many of the adjuncts of the elevator at that time — such as the ropes, hooks, sheaves, roller bearings, etc.

*Brick or Tile in Wooden Structures.* In some cases the hatch or shaft in which the elevator travels is entirely of brickwork, and is, in fact, a brick tube set vertically either in the interior of the building or just outside the wall, with doorways or entrances at each floor. When a line of hatchways situated inside a building, in which the

floors and joists are of wood, are to be protected by a fireproof inclosure, the inclosure must be constructed entirely inside the hatchway — that is to say, no portion of the woodwork may protrude through the inclosure. If of brick, and if the walls are thick enough to admit of it, the joists may enter into and rest on the walls of the hatchway, but in the case of a tile inclosure, the hatchway must be framed so as to be self-supporting, and, of course, must be made large enough to admit the tile work and still leave the shaft of the size required.

*Plaster Type in Wooden Structures.* There is another type of inclosure which is fireproof and which is not nearly so expensive as either of the two just mentioned, viz, the plaster inclosure. This also, in the case of wood floors and joists, must pass entirely inside the hatch. It is made by running 1-inch channel irons vertically from top to bottom inside the hatchway at intervals of about a foot apart, and covering them on both inside and outside the hatchway with wire lath, and plastering on this.

*Construction in Fireproof Buildings.* When a building is of steel construction or partially so — that is to say, if the framing of the hatchways is of steel — these enclosures need not be placed entirely inside the hatchway. In that case, the steel beams where exposed must be covered with either tile or 3 inches of concrete. When a hatchway is of brick, the guideposts, etc., may be bolted directly to the brickwork, but when of tile or plaster, the guides must be fastened to the hatch framing, the fastenings being made long enough to reach through the inclosure.

The inclosures should go clear up through the roof of the building and be covered with a skylight. All door frames must be of iron or steel, and should be set at the time of making the inclosure and be built into it. Provision must be made for the sliding of the doors, which must also be of iron or steel.

**Overhead Screen.** Where the engine is situated below, Figs. 185, 186, and 253, and the cables run thence up to the overhead sheaves, it is customary to place a screen or grating a few inches below the sheaves and extending entirely over the hatchway for a like purpose. In such cases it is essential that the screen be made not only strong in itself but properly supported on small I-beams extending clear across the hatchway, the ends being let into the walls of the hatch-

way. Such screens should always be made strong enough to sustain a load of not less than 75 pounds to the square foot, for, when they are employed, they are always utilized by the attendant as a floor to walk on when oiling or cleaning above.

*Floor for Engine in Penthouse.* When the engine is situated in the penthouse above the hatchway, Fig. 151, Part III of Elevators, and Figs. 206, 207, and 254, it is customary to have a thick floor of planks below the engine supported on heavy I-beams, which are strong enough to carry the engine with its load of car, counterpoise weights, and the burden it is to lift. Such a floor effectually covers and guards the hatchway, but it also excludes the light, so that, where the top of the penthouse is formed by a skylight, it frequently occurs that the floor is made of iron grating which allows the light from the roof of the penthouse to pass down into the hatchway.

**Doors for Shafts.** *Type Opening Vertically.* The shaft doors are of two kinds: those which slide horizontally or sidewise; and those which open vertically. The latter usually open in the middle, one half going up, and the other half down. They slide in guides similar to those in use for sash or window frames, and a chain from each top corner of the upper door passes up and over a small sheave and down to the two top corners of the lower half. In this manner one door balances the other, and they may be moved with comparative ease. When closed, a spring lock holds them securely shut and they cannot ordinarily be opened except from the inside of the car, though they are sometimes made so that they can be opened from the outside also by means of a key.

*Meeker Self-Closing Door.* Sometimes these doors are made so that the upper door weighs a few pounds more than the lower one. In such case, they will not stay open of themselves, the preponderance in weight of the upper door causing it to lower, thus bringing the lower door up to meet it. This is done for the purpose of causing them to be self-closing. To keep them open while loading or unloading, a small catch is placed at the level of the floor, which engages the top of the lower door and is held in position by the car. The car, on leaving the floor, releases the catch, and the doors are at liberty to close, which they do by gravitation. This type of door is called the Meeker door, that being the name of the inventor.

*Door Opening Horizontally.* Doors that slide horizontally are hung usually by their tops to rollers which run on a rail and which sustain the weight of the door, the bottom running in a guide to prevent it from becoming dislodged. They are provided with locks on the inside similar to the Meeker door. Sometimes the upper rail and lower guide are set out of level to make the door self-closing, and a catch is provided to hold the door open while the car is at the floor.

**Semi-automatic Shaft Gate.** *Function.* The semi-automatic gate, as its name indicates, is self-acting in one direction only, viz, that of closing. It is usually made of wood for lightness, and is used to protect openings into an elevator shaft where the Meeker or other heavy doors are heavy and unwieldy and the frequent opening of which would entail unnecessary labor. In such cases these heavy doors are closed at night as a protection against fire, but are kept open daytimes while the elevator is busy, and it is to prevent accidents through unwary persons walking into the elevator shaft that the semi-automatic gate is used.

*Construction.* In Fig. 267, is shown the general appearance and construction of the gate. Where these gates are used they usually set on a ledge or sill projecting into the hatchway inside the inclosure about 3 inches. Each side of the doorway a post is set with a groove on one face for the gate to slide in. These posts extend above the top of the doorway to provide a continuous guide for the gate. One post is made hollow and in it runs an iron weight which is connected to the gate by means of a cord running over two pulleys. The weight is not made as heavy as the gate, and at the lower end there is a notch cut in the weight which, when at its lowest point of travel, engages in a latch which is attached at the lower part of the post or weight box. This latch extends through the side of the post or weight box at $A$, Fig. 267.

*Operation.* The latch is only operative when the elevator car is at the floor or landing, at which time an operating block or cam holds the latch in the position shown in the cut, and then if the weight comes down low enough the latch will catch and hold it. It is only necessary to raise the gate by hand to its highest point to cause the weight to move down to the position where it will engage with the latch. As the weight nearly counterbalances the gate, this is a very simple and easy operation, and the gate being raised remains so as

long as the car is at the landing, thus leaving the opening to the elevator clear. However, as soon as the elevator is started either up or down, the car carries the cam or operating block with it and the latch falls back releasing the weight, and the gate being slightly heavier than the weight closes itself by gravity. The latch is made in two parts. $B$ being separate from $C$ and being pivoted at $D$, the point $A$ can move downward while the cam is in the position shown in the detail, Fig. 267, and thus allow the lower part of the weight to

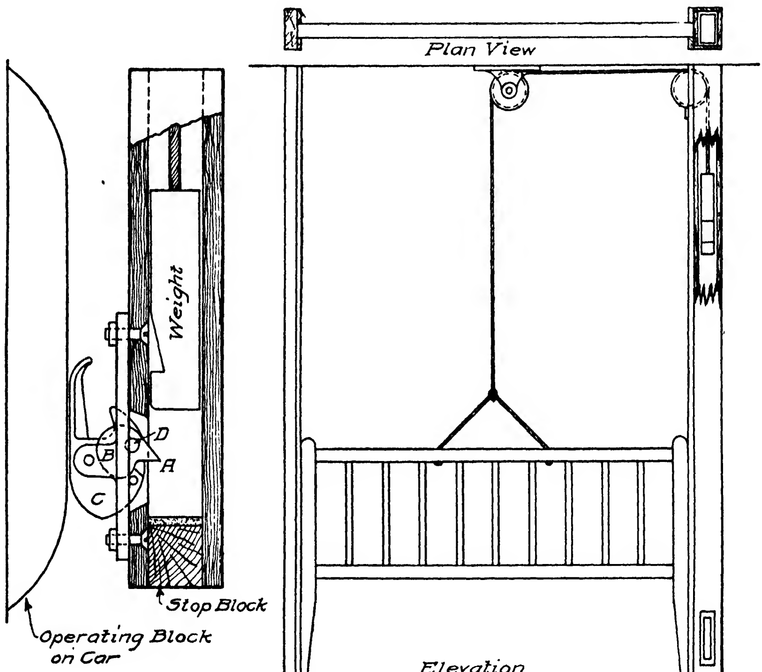


Fig. 267. Semi-Automatic Shaft Gate

pass it. But when the weight attempts to rise in the box, which it will do as soon as the operator who has raised the gate lets go of it, then the cam returns to the position shown at $A$ and locks the weight there.

When the movement of the car carries the cam or operating block away from the back of the latch, it falls out of the perpendicular enough to cause the point $A$ to recede entirely from the inside of the weight box, leaving the weight free to rise and thus allowing the gate

to descend. The weight should always be heavy enough to preclude the possibility of the gate descending rapidly or with force and thus being in itself a source of danger to anyone standing in the doorway.

*Safeguards.* These gates are frequently made to close the opening clear to the floor instead of leaving a space below as shown in Fig. 267, and this is best where it can be done, but usually a lack of head room for the travel of the gate precludes this. In all cases the gate should be made so high that it will be inconvenient for anyone to stand at the gate and rest his arms on the top of it, there being danger of accident from a descending car. Where, through surrounding conditions it is impracticable to have the gate this high, it is customary to hang along the front of the platform from its lower side at intervals of 8 or 10 inches pieces of light chain about 30 inches to 36 inches long, and in case anyone leans over the gate, if a car approaches from above, these chains will strike him in time for him to move out of danger. These appliances, of course, are suitable only for comparatively slow moving freight elevators.

*Entirely Automatic Gate Dangerous.* The full automatic gate is a source of danger and should never be used with any elevator. It opens and shuts whenever the car passes a floor, and, should anyone standing at a landing when the car is approaching lean against or take hold of the gate, he would be liable to get hurt or to be knocked down the hatchway.

**Passenger-Elevator Inclosures.** The preceding description of inclosures and doors applies principally to freight elevators. Passenger elevators are in almost all instances situated and arranged differently. They frequently occur in groups of two or more, and, even when they are isolated, it is a common occurrence to find them located in or close to a stairway. In such cases, it is usual to construct a brick or tile vault or shaft the full height of the building, as before described, isolating the elevators and stairways from the rest of the building, but having landings and entrances at every floor. The passenger-elevator shaft is guarded by an ornamental-iron or steel openwork inclosure.

Of course, many cases occur where the passenger elevator, or two together, are located in a brick or tile hatch entirely isolated from the rest of the building, the only means of communication being through the doorways at each floor, and these doors being of iron or wood

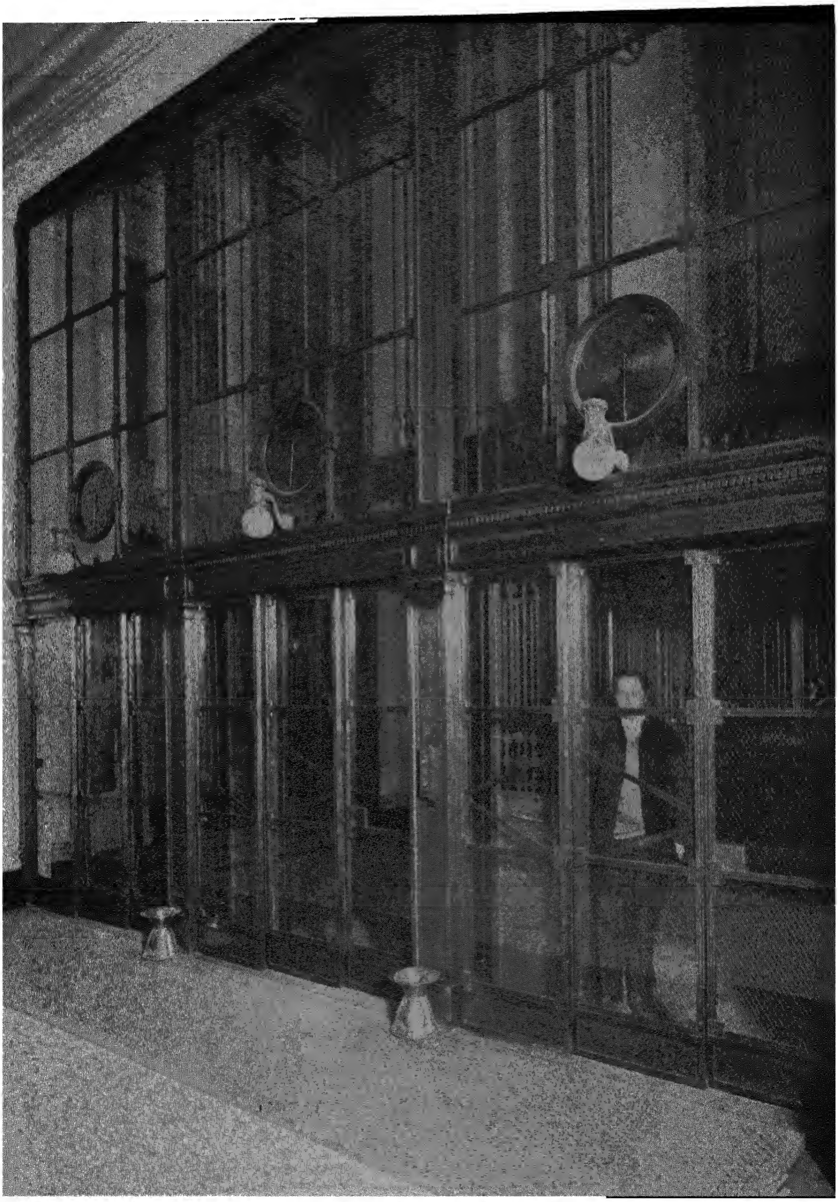

Fig. 268. Arrangement of Fireproof Shaft Inclosure with Signaling Devices
*Courtesy of The Standard-Tyler Company, Chicago*

covered with sheet iron or tin and sliding entirely within the hatchway. Cases like this are not met with so frequently as is the first described arrangement under which groups of six or eight or more

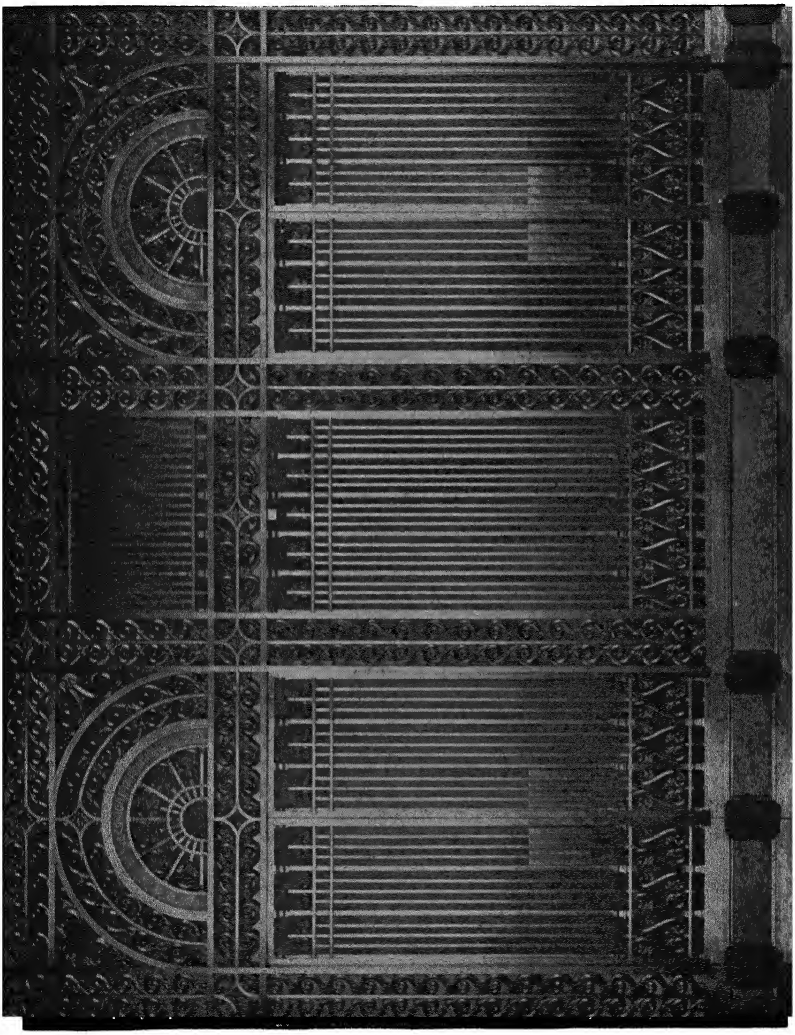

Fig. 269. Ornamental-Iron Shaft Inclosure
*Courtesy of The Standard-Tyler Company, Chicago*

elevators adjacent to one another are to be found all within the same brick or tile shaftway. In such arrangements the ornamental work around the hatchways proper frequently assumes a very imposing character, these inclosures being fitted with lamps over the doorways

to announce to the waiting passenger the approach of the elevator in response to the pressure of the push button of the signaling device,

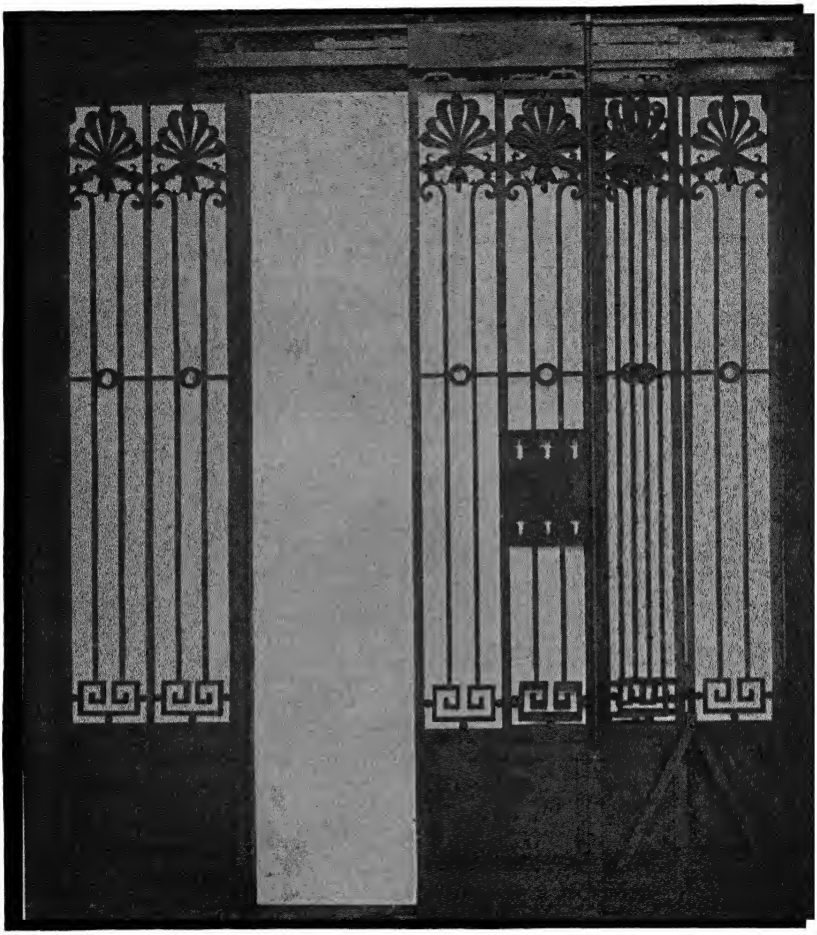

**Fig. 270. Arrangement of Ornamental-Iron Inclosure Gate**
*Courtesy of The Standard-Tyler Company, Chicago*

and with the indicator showing the exact location of each elevator at all times.

In Figs. 268 to 270 are given illustrations of these inclosures with the lamps and indicators, as well as inside views of the inclosures, showing how the doors which close and protect the entrances are hung and secured. These doors are suspended by hangers which

are fitted with double flanged rollers running in antifriction roller bearings. These rollers run on rails set horizontally above the opening in the inclosure, the lower end of the door moving in a groove or track in the threshold. When closed, a lock and catch on the edge of the door holds it shut securely so that it cannot be opened from the outside except by the use of a key; but to one riding in the car the lock is quite accessible to the hand.

### Car Inclosures

**Cabs.** *Construction.* With passenger elevators it has always been the custom to use a cage technically termed a *cab,* Figs. 254 and 271, which entirely surrounds the platform except at the doorway for entrance and exit, the top also being protected by this guard.

The construction and design of cabs differ with the taste and disposition of the purchaser, the range being extremely great from the simple plain wire inclosure to the intricate and elaborate design comprising a grille made of solid and ornamental bars of steel mounted on mahogany paneled skirting with marble baseboard, the canopy and grille highly polished and plated and costing upward of a thousand dollars.

*Use of Seats.* Ideas of convenience differ in different localities. In most large cities, especially in office and public buildings, it is conceded that a seat is an incumbrance, occupying what might otherwise be standing space, and in such places they are never used, but in other localities it is considered that a cab without a seat is incomplete and lacking in convenience. In such places very finely upholstered seats are frequently met with.

**Gates.** In almost every state in the Union laws exist which call for guards or inclosures around all freight elevator cars except at the side where the loading and unloading is done, and where there are more than one of these sides for unloading it is expected that a collapsible gate will be used.

*Collapsible Type.* This is a gate made on the principle of the lazy tongs, and it comprises a number of flat bars of steel laid across one another like the stripes in a plaid. They are pivoted or hinged to one another in such a way that the gate may be extended to close an opening or doorway, it then having the appearance of a number of flat bars laid diagonally crosswise of one another with openings of diamond shape between the bars about 4 inches by 4 inches. The

gate is hung by hangers fitted with grooved rollers which run on the top edge of a flat bar about $\frac{1}{4}$ inch or $\frac{3}{8}$ inch by 2 inches. This bar or

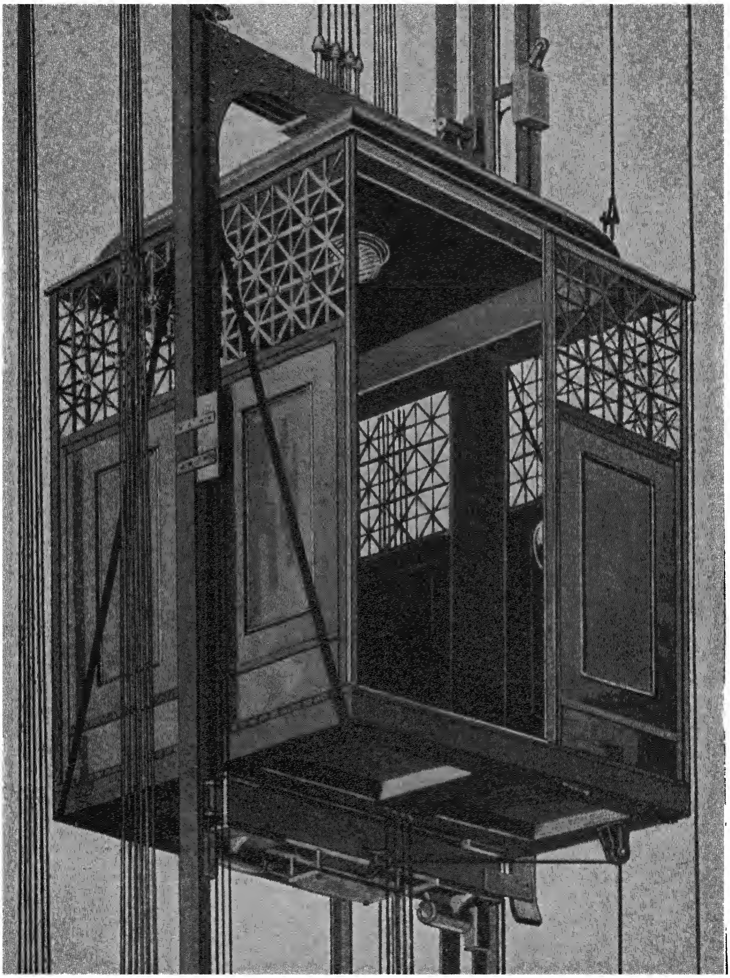

Fig. 271. Details of Arrangement of Passenger-Elevator Car
*Courtesy of The Haughton Elevator and Machine Co., Toledo*

rail is set on edge across the top of the opening, and the gate may be opened by pressing it sidewise, when it will close up or collapse into a space of a few inches, leaving the doorway free or clear for use.

*Circuit-Breaking Arrangement.* These gates are also used in passenger cabs which have more than one opening or doorway, and on electrically operated elevators they are frequently fitted with the electric lock, in which case the opening of the gate opens the operating circuit so that the elevator cannot be started until the gate is closed.

**Car Guards.** These gates, although used as a preventive of accidents, have in themselves a source of danger to fingers if in the way when the gate is being opened, and it is a very prevalent custom for passengers to stick their fingers through the openwork of cabs to hold on to it for the purpose of steadying themselves while the car is in motion. This is a habit that should be avoided, especially where the gates are collapsible gates, for the opening of the gate is very disastrous to any fingers which may be sticking through the apertures at the time.

*Wire Netting.* It is customary on the part of the inclosure and cab maker to attach to inclosures, on the inside at those places which are accessible, and where the counterweight passes, a fine wire netting to prevent people from inserting their fingers and having them injured by the counterpoise weight in passing.

*Wood and Combinations.* Inclosures around freight cars were originally made of wood and for the sake of cheapness are frequently so now, Fig. 225, the guard in such cases being made of wainscoting about $\frac{7}{8}$ inch thick, suitably capped either with a wood cap, or an angle-iron frame. Sometimes the guards are made of wirework; at others, of a combination of wainscoting to a height of about 4 or 5 feet, with wirework above; and oftentimes a wirework guard or screen is placed like a canopy over the whole car to prevent anything falling from above or anyone riding on the car, Fig. 227.

## CAR SIGNALS

**Signaling Device.** The signaling device is an ingenious electrical contrivance, the only outward and visible sign of which is the double push button at each floor and the lamps mentioned. The two push buttons — marked *UP*, and *DOWN* — in one box fastened on the front of the inclosure at a convenient location and proper height serve the entire bank of elevators.

In operation, the prospective passenger on any floor presses the appropriate button indicating which way he desires to go, and then

watches the ground-glass globes, one of which is located above the door of every elevator. Upon the approach of the first elevator going in the direction selected, and while still a story away, the lamp above the door of that particular elevator glows, thus giving notice that the elevator is arriving at that floor. At the same time a lamp in the cab located just below the level of the operator's face glows in a similar manner as a notice to him to stop there.

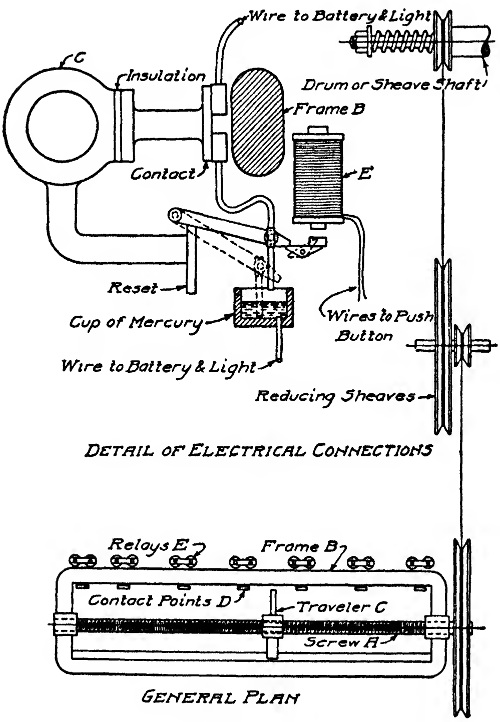


Fig. 272. Diagram of Signal Lighting Device

*Mechanism.* Fig. 272 is a diagram which shows how these results are attained. The mechanism comprises a long screw *A* revolving in a frame *B*, the screw has a traveler or nut *C* running from one end of the screw to the other as it revolves, and in its passage it passes a number of contact points *D* which represent the different stopping places of the elevator. These contact pieces are connected by wires to a battery or to a weak electric current derived from the lighting plant or other service of the building so as to form two separate circuits for each floor, one for up the other for the down motion of the car. In each of these circuits are a number of relays *E* corresponding to the different landings, each circuit and each set of relays being insulated from each other. Inserted in these circuits are the push buttons at the different floors, the *UP* buttons being in one circuit, the *DOWN* in the other, and they are so arranged that the circuits are always open except when a button is pressed.

*Typical Operation.* The pressing of a button at any floor closes that particular circuit. Suppose it is the *UP* button, then that particular circuit is closed, and this actuates the relay on that particular circuit, causing it to attract a small iron bar placed across the poles of the relay but a short distance from it. This bar, in rising to the poles of the magnet, trips a catch which allows the end of a wire connected with the circuit to drop into a small cup of mercury, thereby closing the circuit for the time being independently of the push button, so that, if the finger is removed from the latter, the circuit remains closed. Then, when the traveling nut on the screw arrives at the contact pieces corresponding to that floor, it cuts two lamps into circuit, one in the cab, and one at the floor or story where the push button was pressed. This traveling nut is so adjusted in relation to the contacts in the frame that they meet when the car is fully a story in advance of the floor at which the button was pressed, and the lamps glow that much in advance of the stopping place, thus giving both the prospective passenger and the operator timely warning. The lamps continue to glow till the car, after having stopped and taken on the passenger, leaves the floor, when the circuit is broken and the lamps become dead again. This is brought about by the traveling nut on the screw operating a rocker arm which lifts the wire out of the cup of mercury and resets it again as before, thus opening the circuit.

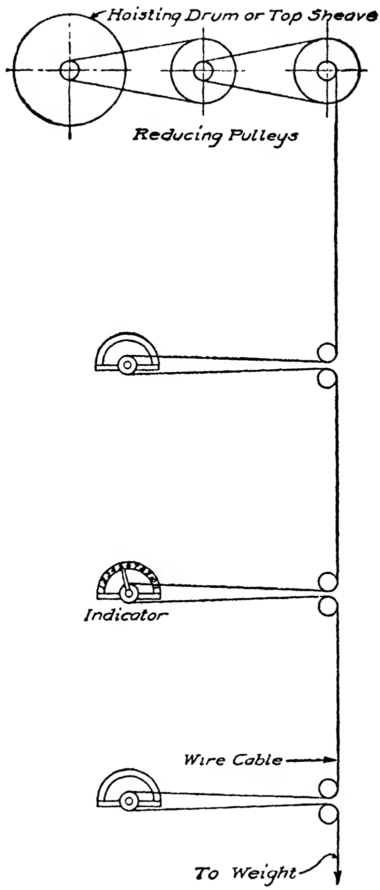

Fig. 273. Diagrammatic Arrangement of Mechanical Indicator

**Mechanical Indicator.** Another convenience in connection with the ornamental inclosure is the mechanical indicator which shows the

location of an elevator relative to the floor on which the passenger waits. It comprises a hand or pointer, in some cases moving vertically along a scale, and in others revolving around a circular or semicircular face like a huge dial, which indicates the position of the car. As shown diagrammatically in Fig. 273, this is brought about by differential gearing, usually of the link-belt and sprocket-wheel type, which is driven either from the drum shaft of the engine or from the shaft of one of the overhead sheaves and is geared down so that the hand or pointer travels only about $\frac{1}{40}$ as far or as fast as the car, or in many cases $\frac{1}{60}$ of the travel of the car. If the car is making a run of, say, 150 feet, the travel of the pointer is 4 feet or less, according to the ratio of the gearing.

The space traveled by the pointer being marked at intervals by numerals corresponding to the number of the floor, these figures are set at certain distances apart on the dial corresponding to the heights of the several stories — but, of course, relatively less — so that, as the car passes or stops at each respective landing, the pointer indicates its real position at any time.

**Annunciators and Lights.** *Usual Annunciator Arrangement.* In passenger cabs is commonly found a small box about 4 inches wide by anywhere from 9 to 24 inches in length which is fastened to one of the sides of the cab in such a position as to be seen readily by the operator. The box is sometimes of wood, at others, of enameled or polished metal. The face is of glass blackened on the inside, except that at regular intervals along the face in a vertical line a row of circular spaces are left clear. Inside the box behind the glass and between the openings are arranged a series of small electromagnets, which are connected by wiring to a battery located at some convenient point in the hatchway. The wiring is continued to push buttons at every floor or landing, and is arranged exactly similar to the wiring for a doorbell. The number of openings in the box in the cab corresponds to the number of floors or landings the elevator has, as do also the number of relays or electromagnets. Each magnet is wired directly to its appropriate button at the landing, the button being set into the inclosure surrounding the hatchway, and all the magnets being connected to the battery.

*Annunciator Operation.* All the circuits are open and remain so until the push button at any floor is pressed. This closes the cir-

cuit to the corresponding electromagnet which at once attracts a small soft bar of iron set across the poles but a short distance away. The rising of this bar to the poles of the magnet releases a catch which allows a small disk called a target to drop in front of the opening. This target has a number on it corresponding to the number of the floor at which the button was pressed. The pressing of the button also places in circuit a small buzzer or bell, as the case may be, which is also located in the box. This bell calls the attention of the operator, who can, at a glance, see which floor the call emanates from. A knob at the bottom end of the box is attached to a rod passing up through the entire length of the box. At each target an arm projects from the rod, and by pushing up the knob the operator restores to their original places any of the targets which have dropped to the visible position. Sometimes this synchronizing of the targets is done electrically by using another electromagnet to operate the rod, the push button being set at a point close to the operator and connected to the magnet by concealed wiring.

It is perhaps needless to say that the apparatus just described is called an annunciator because it announces the floor at which a passenger is waiting. It was originally introduced for use in hotels as a call for the various rooms to the office and took the place of the noisy gong which more than a generation ago was in use. Its application to the elevator is merely an adaptation, but many improvements have been introduced since its first introduction.

*Annunciator Double Targets.* For instance, in many cases an annunciator box with a double set of targets is used — one vertical row being for *UP* and the other for *DOWN* — and double buttons marked the same way are placed at each landing. The passenger presses the button corresponding with his desired destination so that the operator, having his attention called to the annunciator, knows at once whether the person calling desires to ascend or descend, and governs himself according to his location. Where two elevators are running in the same shaft, they are cross-connected so that a call is announced in both cabs simultaneously — only one set of buttons being used for both. The cab which is in the proper position responds, for, where elevators are run in the manner first described, it is always arranged that one is up when the other is below.

*Annunciator Lamps.* Also, in many instances the targets in the annunciator box are dispensed with, and instead there is set opposite each opening in the glass face a tiny incandescent lamp, which is made to glow and remain glowing when the button is pressed; the number of the floor being put on the glass face of the box, the glowing lamp indicates the floor from which the call comes.

*Annunciator Cable Connection.* The manner of connecting to the moving cab the wiring which operates this annunciator is simple. All these wires — as well as the lighting and control conductors — are led to one point in the hatch about midway of the run, that is to say, between the top and bottom landings, Figs. 254 and 271. At this point there is attached a flexible electric cable which comprises a number of insulated wires, one for each electromagnet, and one for the battery. These wires are all bound together and covered with a suitable covering of mohair. The cable is long enough to reach down from this point in the hatchway just described to the lowest landing, or up to the highest, and still have 10 or 12 feet to spare. It then is attached to the bottom of the cab, leaving plenty of slack, and from there is run along the outside of the cab to the annunciator box where the wires are connected.

*Cab Lighting Connections.* A similar but separate cable having only two wires is used in the same manner to produce light in the cab, the cable being connected to a light outlet in the hatch and thence to the cab. One wire is run directly to an incandescent lamp set in the dome of the cab, and is usually covered by a glass globe; the other wire is run also to the lamp, but is carried to a point near the operator where a snap switch is introduced, thus enabling the operator to turn on or off the light.

In some large cabs two or more lights are used in a cluster — in such cases there being in the cable two wires for each light, with a snap switch for each inserted in the circuits — but such arrangements are all matters of taste on the part of the owner or architect.

## ELECTRICAL ELEVATOR CONTROL

**Control-Circuit Devices.** *Automatic Centering Operating Switch.* Right here, it may be well to describe another safety feature which is used frequently in connection with elevators which are operated by electric current. One feature of the operating or master switch used in

the cab for stopping and starting the elevator is that if the operator's hand is removed from the handle at any time the switch at once returns to the stop position. This is to prevent accidents which might be caused through inattention or inadvertence.

The reader who has perused the description of the electric elevator will be aware of the fact that the current which is used for controlling is not the same as that which supplies the power to run the motor. It may be taken, and usually is, from the same source, but it is converted to a lower voltage—never higher than 110 volts, and may be less—and but 2 or 3 amperes are required in operation, the current being used simply to actuate the solenoids which open and close the main line switches.

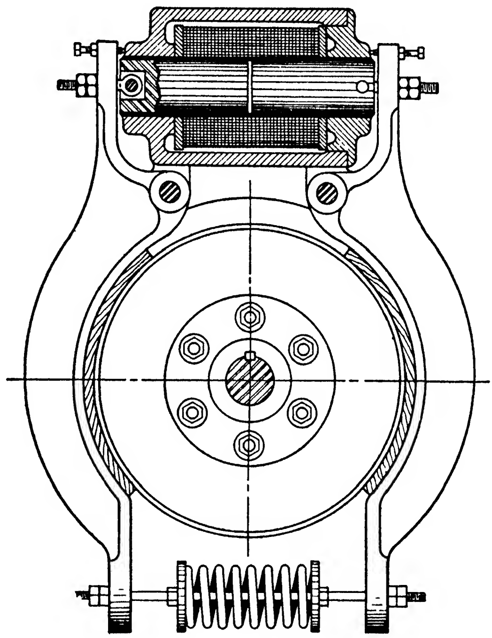
Fig. 274 Electrical Solenoid Brake

*Door Operated Circuit Breaker.* Sometimes this circuit is wired up the entire length of the hatchway and is so arranged that the opening of a door anywhere in the run opens this circuit, which causes the elevator to stop and prevents its being started again until the door is closed. This feature makes it impossible to start the elevator while the door is open. Hence the car cannot leave the floor until the door is safely closed. The manner in which this is brought about has been fully described previously under the head of Push-Button Control in Part III, Electric Elevators.

**Main Operating Circuit Devices.** *Emergency Switch.* Another feature in this connection is the emergency switch, which is one placed in the cab in the case which holds the operating or master

switch, by means of which the movements of the elevator are controlled. The emergency switch opens the main circuit, so that the electric brake is applied at once, bringing the elevator to a stand immediately, and at the same time making it impossible to operate the elevator again until it has been closed.

*Use of Solenoid Brake.* This appliance has been described under the head of Controllers in Part III, Electric Elevators, but a very brief recapitulation will not be out of place here. A powerful brake applied by strong springs is used, and it is released by an equally powerful solenoid, Fig. 274. The opening of the electric circuit by any cause allows the springs to apply the brake at once, and as the solenoid is in the same circuit that operates the motor, it can readily be seen that it must be very efficient and reliable.

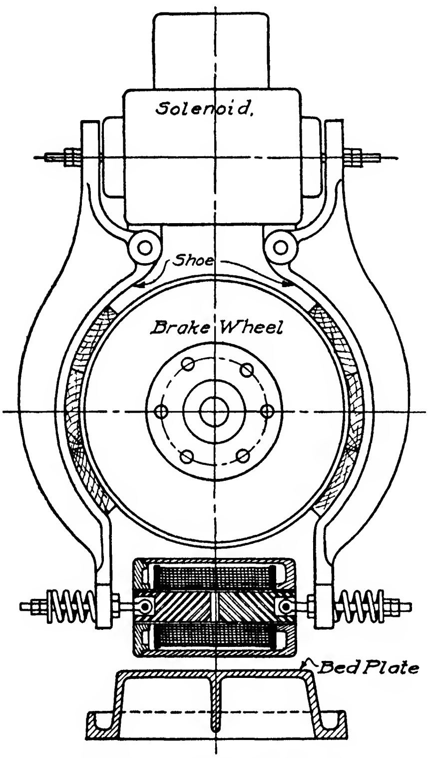

Fig. 275. Extra-Squeeze Brake

*Extra-Squeeze Brake.* Supplementing the main solenoid, another is sometimes used which is applied in such a manner that when energized it acts to help the springs which apply the brake, Fig. 275. This solenoid is on a separate circuit which is never closed except when the car goes beyond the floor some inches at either end of the run. Should this occur through excessive momentum, the car strikes the arm of a switch closing the circuit, which causes the solenoid to give the brake an extra squeeze.

*Drum Brake.* Another form of the solenoid brake is sometimes applied to the drum also, Fig. 276.

**Car Locking Device.** *Extra Heavy Load Requirements.* Electric elevators frequently are made to run at high speeds for ordinary

traffic, but also are arranged to lift heavy loads at a slower speed. Where a bank of three or more elevators is installed in an office building, it is found very convenient to have one of them arranged in this way, so that, while the elevator under ordinary circumstances is used for the regular traffic, it can on occasion be converted into a slow running elevator having a lifting capacity of twice or three times what it has at the higher speed. This is found to be convenient for moving safes and other heavy objects. The extra power is sometimes obtained by the use of compound gearing, which can be thrown into

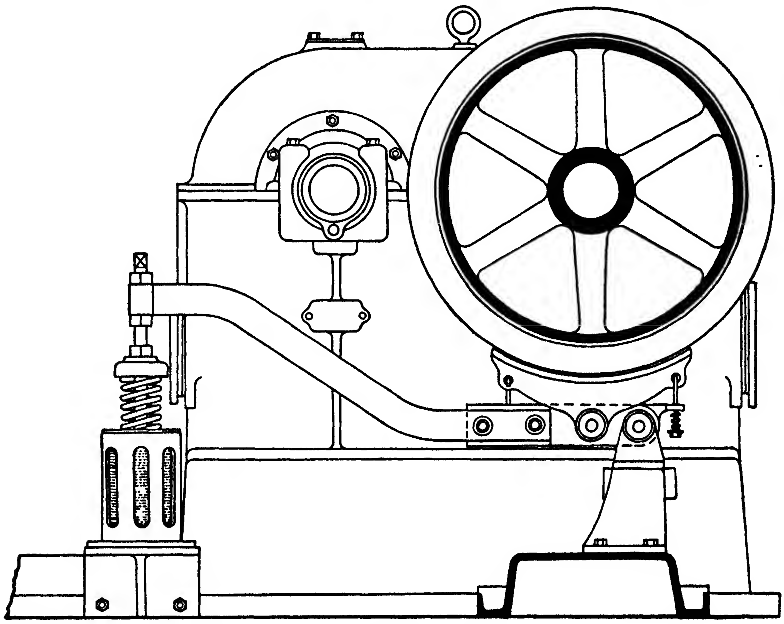

Fig. 276. Solenoid Brake Applied to Drum

or out of engagement at pleasure. In other cases, where the motor used is a 2- or 3-speed direct-current motor having extra heavy fields, the load may be lifted by using the first speed only, which runs the motor with the fields strengthened to the fullest extent.

*Locking Operation.* When an elevator arranged in this manner is used for lifting heavy loads, it is customary to equip it with heavy bolts or bars beneath the platform, which, by means of a lever, may be made to engage in catches attached to the guides. This is done by operating the lever which slides these bars till they engage with the

safety catches, then, when the load is on the car, the elevator is lifted an inch or two and the bars are withdrawn. Upon the arrival of the

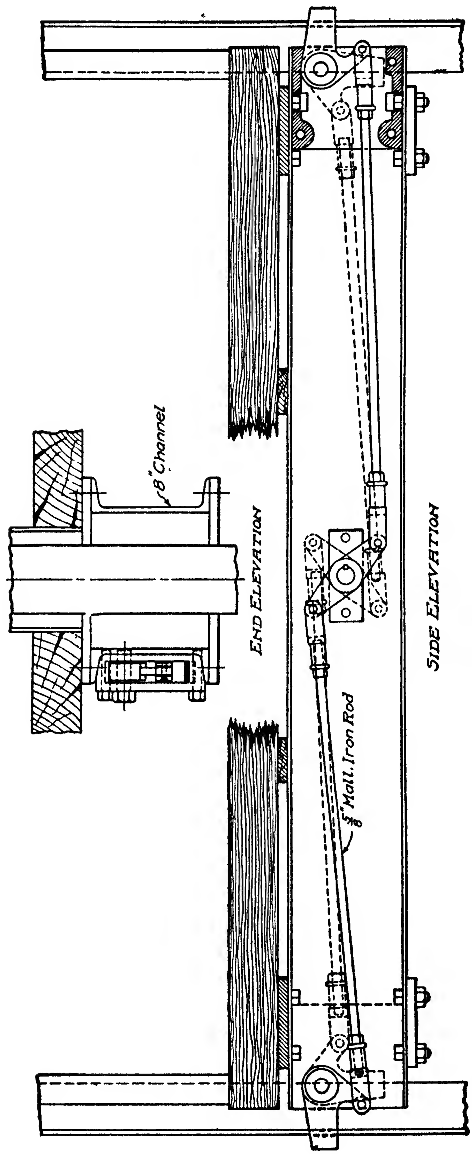


Fig. 277. Car Locking Device for Traction Elevator

car at the landing where it is to be unloaded, the catches are again thrown before unloading. Such an arrangement is shown in Fig. 277.

## CUSHIONING DEVICES

**Air Cushion.** *Characteristics.* Probably the most unique and most efficient safety ever invented to save a falling car is the air cushion. That is to say, it is the most efficient when properly constructed and kept in order. But here its merits end, for it is difficult to construct and more so to keep in perfect working condition, and a safety must be reliable above all things; it must not be prone to get out of order, and the air cushion has all these faults.

*Cylinder on Car.* It was first thought of by an engineer named Gray, of Cincinnati, who patented it in 1879. His claims were a bell or cylinder open at one end, with the open end down, and attached by its upper end to the underside of the platform. In the pit at the bottom of the run was permanently fixed a piston which fitted loosely in the cylinder, which was made about 6 feet long. It was expected that if the car fell, the air contained in the cylinder, being confined and escaping slowly, would eliminate the shock consequent to the fall. While the idea was good, his application of it was hardly practical except for very low buildings.

*Shaft Cushioning.* Six months later a patent was obtained for a similar but differently constructed device by a man named Ellithorpe, of Chicago. His idea was to utilize the lower story of the elevator shaft for the cushion and to make the platform act as the piston by putting rubber strips around its edges, until it filled the opening or space between the car and sides of the hatchway to within about $\frac{3}{8}$ inch of the sides of hatch. Of course, the sides of this chamber formed by the lower story of the hatch must be straight, and any door opening into it must be fitted air tight and always be closed, and the walls of the chamber must be very strong. The weak points of the arrangement being the tendency toward leaving the door ajar, or the disarrangement of the rubber strips, either of which would destroy its effectiveness by allowing the too free escape of the air from below the platform.

At the time this device was first used there were very few buildings higher than 6 or 8 stories, but since the introduction of 10, 15, and 20 or more stories the use of the lower story alone for the air chamber is not sufficient, 18 to 36 feet being necessary for effectiveness, and in the case of a building like the Woolworth of New York City, an air chamber of not less than 50 feet would be required.

It will be understood that the efficiency of the air cushion depends on the retardation of the escape of the air from the chamber while the platform is descending into it with great speed and force. On the other hand, if the platform fits the chamber too closely, a severe shock will be experienced as soon as the car enters it, and inversely the existence of either of the conditions mentioned above which allow too free an escape of air will destroy its usefulness. Hence, it is not in general favor.

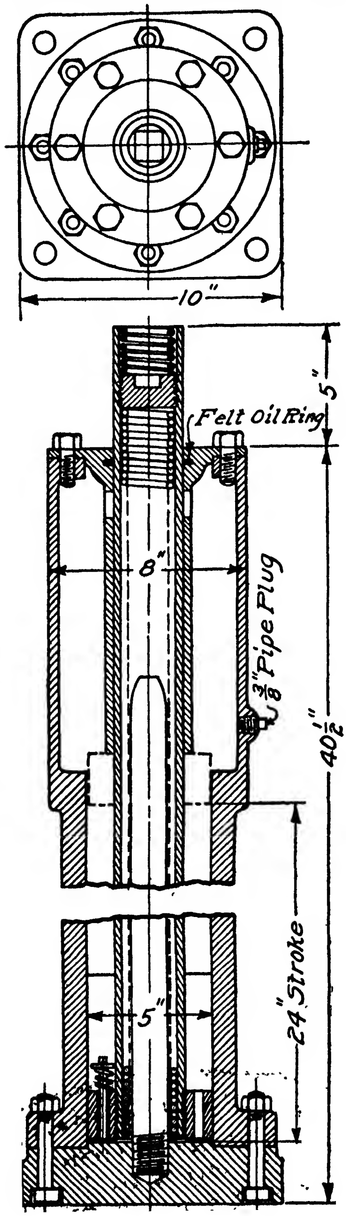

Fig. 278. Warner Oil Buffer

**Buffers and Bumpers.** *Spring Buffer.* On all traction elevators it is customary to use either spring or oil buffers. The spring buffer is simply a very long spiral spring inclosed in tubes which telescope into one another and which are set in the pit below the lowest landing, one under the car and one below the counterpoise weight. They are made with a stroke of from 2 to 3 feet, and an extra deep pit is required to make room for them.

*Oil Buffer.* The oil buffer, Figs. 166 and 167, Part III of Elevators, and Fig. 278, is a cylinder hung from the bottom of the car, and similarly from the weight. It is kept filled with oil and is suspended in such a way that it can travel upward vertically if it meets with an obstruction. Directly above it and rigidly attached to the car or weight bottoms is a plunger which fits its bore exactly and which has grooves cut in it to allow the passage of

the oil. If either the car or weight go below a certain point, the end of the cylinder strikes an obstruction such as a pier or beam set there for the purpose. The continued descent of car or weight, as the case may be, causes the cylinder to be forced up on the plunger, and as the only escape for the oil is through the grooves before mentioned, this retardation opposes the descent of car or weight, bringing it to a gradual rest.

*Combination Buffer for High-Speed Elevators.* An independent or stationary oil buffer is also used on high-speed elevators which is a combination of the long-stroke spring buffer and the oil buffer just described, the oil being used as the retarder and the springs to restore the plunger to its normal position after use. This type of buffer requires a deep pit and is set stationary on a pier below the car and weight. Its advantage is the long stroke.

These appliances are only called into requisition by the slipping of the brake.

*Bumper.* Bumpers are simply springs of either steel or rubber for the car to rest upon at the lower landing, and are used only with drum machines, as in Fig. 207.

**Guide Oilers.** These are hardly safeties as the term is generally applied, but, as they conduce to the general efficiency of the elevator, are mentioned here. They are simply reservoirs of oil which are set on the upper guide shoes of both car and counterpoise weights, and by means of wicks feed oil to the guides.

## CONCLUSION

**Intelligent Handling Imperative.** The devices described comprise the principal safeties used in the equipment of elevators. It will be admitted that they are quite numerous, but with all of them there is none so important nor so effective as constant and eternal vigilance in efficient and intelligent frequent inspection, and care in handling the machine.

# REVIEW QUESTIONS

# REVIEW QUESTIONS

ON THE SUBJECT OF

# ELEVATORS

## PART I

1. Draw a diagram of an elevator showing use of served-rope hoist.

2. What is a striker?

3. Describe a drum governor.

4. Sketch and describe the box locking device for a dumb-waiter of the domestic type.

5. Give a description of the general construction of the four-chain type of basement elevator.

6. Enumerate the differences between the modern type and the sling-type hand elevators.

7. Describe the steel-plug type of thrust device. Why is such a device necessary?

8. Make sketch showing the details of the cylindrical-cam belt shipper.

9. What are the requirements of a steam engine for belt-driven elevator service?

10. How is the four-chain type of basement elevator operated?

11. What are the best worm and gear proportions?

12. Sketch the steel band type of elevator brake.

13. Describe the method used to make a dumb-waiter of the heavy-service type stand wherever it is left.

14. What is a centrifugal safety device?

15. What is the office of safety dogs?

16. What is the hob?

17. A load of 2500 pounds is lifted by a worm-gear elevator; the drum is 24 inches in diameter and the worm gear 20 inches at the pitch line. Calculate the end thrust of the worm shaft.

18. By what means is the reversible type of motor operated?

19. Why did the efforts to use on the worm a small thread angle fail to give satisfactory results?

20. What is the most efficient thread angle?

21. What is an outrigger?

22. Why was it necessary to adopt the center lift for elevators?

23. What is the object of a counterpoise weight?

24. Describe a wagon and carriage lift.

25. What is the purpose of a dumb-waiter?

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# REVIEW QUESTIONS

ON THE SUBJECT OF

# ELEVATORS

## PART II

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1. Describe the Crane limit valve.

2. Give a general description of the direct-plunger hydraulic elevator.

3. What is the object of the pilot valve?

4. What difficulties were experienced in using high pressure with hydraulic elevators?

5. Make sketch of the shock absorber and explain its principle.

6. Give the working principles of the Otis Tufts valve.

7. Describe the Armstrong hydraulic crane and state what resulted from its development.

8. What is the object of pressure tanks in a hydraulic elevator system?

9. Make a diagram of the Hale standard hydraulic elevator and explain its working principles.

10. On what principles does the Hale valve operate?

11. What are the disadvantages of so-called pull machines?

12. Sketch an operating-lever device with extra sheaves.

13. Explain the action of an accumulator in connection with the hydraulic elevator.

14. Describe the sidewalk lift with telescopic guides.

15. What is the purpose of a traveling stay?

16. Make sketch of an Otis steam valve.

17. What gear speed was attainable after the introduction of the internal spur gear.

18. Sketch details of Armstrong operating valve.

19. What were the causes for constructing push machines?

20. Make a diagram showing the method of using counterbalancing chains.

21. What are the chief essentials of a speed regulator?

22. How much pressure is used in compression tanks?

23. What is the office of stop buttons?

24. Explain the working of a siphon relief.

25. Why was the Armstrong three-cylinder machine not successful?

26. Make a sketch of the early New England type of low-pressure hydraulic elevators.

27. Give a description of the two-way valve.

28. What improvement did Witte introduce in the operation of the high-pressure elevator?

# REVIEW QUESTIONS

ON THE SUBJECT OF

# ELEVATORS

## PART III

1. Give a description of an early type of elevator installation.

2. What are the requirements for motors?

3. Describe an interpole motor.

4. What difficulties are encountered in using alternating-current motors for elevator work?

5. What are the functions of a controller?

6. In what particular points does an elevator controller differ from an ordinary motor starter?

7. Give diagram and description of a semi-automatic control for compound-wound dynamos.

8. Describe the mechanically operated type of alternating-current controller.

9. Give wiring diagram for a three-phase elevator equipment with mechanical control.

10. Give sketch of a single-speed full-magnet controller with explanations of its working.

11. Analyze briefly the control circuits in an elevator using the push-button controller with alternating-current motor.

12. Discuss the points which have to be considered in regard to the location of an elevator engine.

13. What are the advantages claimed for the tandem-geared engine?

14. In what case are double-threaded worms employed?

# REVIEW QUESTIONS

ON THE SUBJECT OF

# ELEVATORS

## PART IV

1. How do you determine the distribution of overhead loading?

2. Describe fully the strains on the teeth and rims of spur and worm gears and state how they differ.

3. What are the essential elements in the proper construction of an elevator car?

4. What kind of strain is a hub key subjected to and how do you find the proper proportions for this member?

5. What advantages do cars with guides at the sides have over those with guides at the corner?

6. Describe the various methods of scoring hoisting drums, set sheaves, and vibrating sheaves, and state why they differ.

7. What is the function of the truss rod in a platform and in what type of framing is it used?

8. What are the principal requisites for a top sheave box or bearing and how do you arrive at the correct proportions for same?

9. What is the use of a counterpoise weight?

10. How do you find the proper diameter of sheave to use with a given diameter of cable?

11. Classify and describe the principal stresses to which elevator machinery is usually subjected.

12. Describe how to shackle a cable and state what precautions are to be taken in doing it.

13. What are the chemical elements found in cast iron which unfit it for elevator service and how do they severally affect it?

14. Describe the several types of guideposts and guides in use.

15. What precautions are to be taken in designing sheaves and drums and give the reasons therefor.

16. How are guideposts and guides set and lined up? Describe the method of doing this.

17. State briefly the different stresses to which horizontal drum and sheave shafts are subjected and the methods of determining each separately.

18. What are the advantages gained by setting the winding engine overhead instead of locating it at the bottom of the runway?

19. Describe the materials entering into the construction of wire hoisting cables, method of manufacture, care, conditions which tend to weaken them, and how to properly inspect them.

20. What is the meaning of the terms dead load, live load, and impact, and how are they determined?

21. What is the rule for finding the safe working load for a given diameter of cable?

22. What is the most essential feature in selecting overhead beams?

23. Give a brief description of several kinds of cable fastenings and their individual uses.

24. Describe the different types of platform construction.

25. Give a brief description of each of the various methods of fastening guideposts and guides.

26. What proportion of the load do the side rails, front, back, and cross rails usually carry?

27. In what particular essentials does a traction machine differ from a drum type of engine? Describe each type briefly.

28. Describe separately stiles, side braces, main crossbeams, and truss beams, and give their uses.

29. Give a comparative analysis of the requirements for overhead beams in each arrangement.

30. How do you determine the proportionate amount of car weight and drum weight?

31. Describe the compensator and its uses.

# REVIEW QUESTIONS

ON THE SUBJECT OF

# ELEVATORS

## PART V

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1. Describe the signaling device. Also the mechanical indicator.

2. Give in detail the difference between a pull and thrust hydraulic.

3. What is an annunciator? Describe its use and operation.

4. Describe the construction and action of the grease cup used in lubricating pistons and cylinders.

5. Describe the car-locking device and state its uses.

6. What constitutes an electric horsepower?

7. What is an air cushion?

8. What are the essentials of a good belt for elevator service?

9. Describe Meeker doors and semiautomatic gates and the conditions calling for each kind.

10. How do you calculate the electrical horsepower required to lift a given load at a given speed?

11. Describe methods and give examples of finding the following: Proper thickness of walls of hydraulic cylinders, die and pressure being given; proportions of operating valves; and thickness of cylinder heads.

12. What essential feature should predominate in the selection of a motor for elevator service?

13. What are the essential features and conditions to be observed in piping hydraulic engines?

14. How do you calculate the weight of water in a pipe or in a tank?

15. What is generally conceded to be a man's effective power in pulling a hand rope or in turning a crank?

16. Describe the various types of platform safeties and how operated.

17. What is the difference between a kilowatt and a horse-power?

18. How is the overhead electric engine supported?

19. How do you arrive at the proper width of a belt for elevator service?

20. Describe the various kinds of platform inclosures including cabs.

21. What is the difference between a direct-connected and a single belt elevator?

22. Describe the various types of electric brakes.

23. How do you estimate the quantity of water used by a hydraulic elevator in making a round trip?

24. Describe the car switch and the emergency switch.

25. What advantage is there in counterbalancing the cables? Describe how it is done.

26. Describe the different kinds of bumpers and buffers.

27. What is the office of the slack cable stop? Describe its operation.

# INDEX

# INDEX

*The page numbers of this volume will be found at the bottom of the pages; the numbers at the top refer only to the section.*



*Note.—For page numbers see foot of pages.*

*Note.—For page numbers see foot of pages.*

*Note.—For page numbers see foot of pages.*

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